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DESIGN · PRODUCTION · MATERIALS

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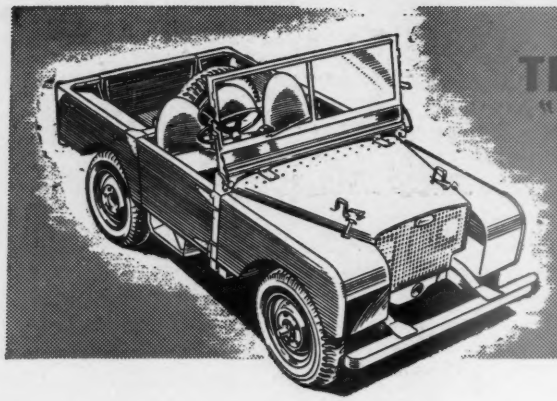

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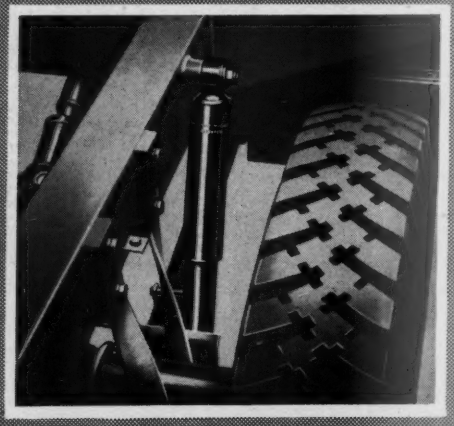

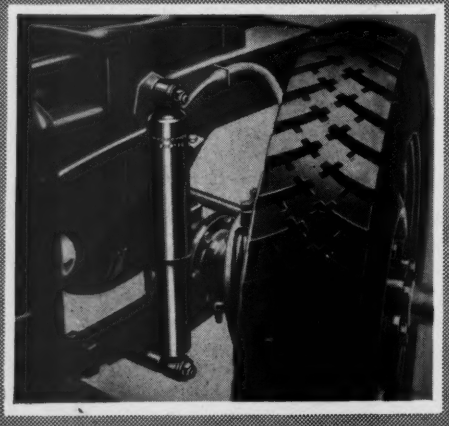

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AUTOMOBILE ENGINEER

Design, Materials, Production Methods, and Works Equipment

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Power Steering

IT is probably twenty-five years or more since designers first began thinking about power steering. Certainly it seems rather odd that on heavy vehicles the purely manual gear should still be so general. Much of the reason for this lies in the remarkable achievements of the component manufacturers. Proprietary gears are now relatively frictionless, so reliable, and low in cost that the anticipated development has been delayed. The additional cost and complication of an assisted mechanism naturally constitute a prime objection. There is also the fact that both design and execution of such mechanism must be of first quality. Failure in even the smallest degree might engender fatal consequences. There can therefore be no risky compromises to simplify manufacture or reduce cost.

A mechanical advantage of something in the region of four times the manual effort would probably be a suitable compromise, depending on the vehicle. The design of effective yet foolproof power steering units is by no means easy. All the components from the road wheels to the point in the assembly where the power is applied must be stressed to withstand greatly increased loading, although it may be possible to limit this by means of relief valves. A point that will need watching is the stressing of the ball pins on many private cars. It would seem desirable in any design to omit all lock-stops on the swivels and limit the travel within the power mechanism itself. Attention must also be paid to the possibility of detrimental effect on thin-walled tyres. Screwing up and puckering action may well be caused by power-assisted movement of the steering swivels through considerable angles with the vehicle at rest. The effect of a road wheel contacting the kerb during the initial turn away from a parked position could produce considerable stresses and lead to distortion of the rim.

Physical Effort

The physical effort of steering a modern bus, with a front axle weight of over four tons, should not necessarily be accepted as inevitable. Such vehicles have frequently to be put on nearly full lock at low speeds when pulling away from behind a stationary vehicle. Servo brakes have long since become standard and power operation of self-changing gear pedals is an accepted feature. Even the hand brake is receiving attention in some quarters. The large ground contact of the modern low-pressure tyre makes parking a real effort to many elderly private motorists and generally induces the designer to provide a larger reduction in the steering gear than is desirable for responsive handling on the open road.

Public safety and traffic congestion are to some degree involved in the matter of easy steering. With a high-ratio steering it is difficult to effect a rapid change of direction in an emergency. Inability to apply full lock quickly when pulling away from rest often means that more lock has, in the end, to be used. The vehicle swings further out into the traffic stream than would be necessary if power assistance were available. With power assistance, lower ratios could be used. The ordinary motions of steering could be made without having to shift the hold of the steering wheel.

Obviously, the top-heavy double-deck 'bus should not have the high speed, sensitive response of the sports car. The long wheel base of a typical 'bus type vehicle, does however demand a relatively bigger momentary steering angle for a given angular deviation or curve of given radius. Accidents that could be avoided, without risk of overturning the vehicle, have occurred because the driver was not able to apply the required lock sufficiently quickly. Alternatively, 'buses have been overturned because the steering was too slow. This necessitates taking a second hold so that eventually more lock is applied than was intended and recovery could not be made quickly enough. While these considerations do not apply with equal force to the private car, yet many urban delivery vans are based on standard private car chassis.

Size of Vehicle

Any approach to the problem must of course be appropriate to the weight of the vehicle. A 'bus will probably have hydraulic steering, possibly of the type employing a differential operating piston and a two-port control valve. Such a gear is locked against back-kick and will presumably be less subject to "chatter" on a rough road than the more common four-port arrangements. Here, the road can react on the steering wheel and call for power application, when simple holding is all that is required.

Whatever the type of power steering employed, it would seem that the hand-operated mechanism and steering linkage can be of the highest attainable efficiency. "Back-kick" is dealt with in one way or another by the hydraulic

mechanism. Extremely sensitive steering should therefore be possible and only the lightest touch will be required to straighten out after negotiating a curve. Some types of hydraulic steering will not however recover of their own accord.

High efficiency, with a ball-bearing nut or equivalent mechanism and roller-bearing swivels may make it possible to have reasonably safe steering on the open road in spite of momentary failure of the power by accidental stoppage of the engine. Coasting of a power-steered 'bus would appear to be excluded unless operation is by an accumulator served by both engine- and transmission-driven pumps. With hydraulic mechanisms on private cars, there will probably be a preference for those systems giving a free neutral position. All ports are open and it is possible for the road to be "felt" and for the car to straighten out without assistance from the driver.

Sports Type Cars

Experienced drivers may have a preference for a high-geared steering of high efficiency with pre-loaded centring stops on the control valve. These would be so arranged that normal light steering effort is transmitted from the steering wheel without the "backlash" necessary to operate the control valve. On the open road the car would be steered direct, at high box-efficiency. Any moderate "back-kick" would be adequately damped by the hydraulic operating cylinder circulating the fluid through its open, but suitably restricted, ports.

It might well be found that, on a moderate-sized vehicle intended to appeal to the enthusiast, the combination of a high-efficiency steering gear with hydraulic damping but no added power, gave a very satisfactory compromise. On large cars, the power will be primarily employed to overcome the parking and low-speed disabilities of the modern low-pressure tyre. These tyres make a notable contribution to comfort, but do not make for pleasant steering.

In the search for a simple mechanism, vacuum operation will doubtless be investigated. Here, the preference will probably be for the pre-loaded centralizing mechanism already envisaged but made even more stiff. With a high-geared steering a good deal of driving would be done without calling on the vacuum at all. Since in this instance the operating medium is readily compressible, response will tend to be somewhat sluggish. Alternately, a wasteful amount of "vacuum" will be needed, calling for a large pump additional to the manifold depression, which will not always be available.

Compressed-air operation is not, of course, to be excluded. It will probably find its chief application on cross-country vehicles, excavator's lorries and the like, where delicate control is not essential but brute force is at times required. Air consumption can be appreciably reduced by a system developed by a well-known English firm under which the control valve as it centralizes, connects the two ends of the operating cylinder. This equalizes the pressure without exhausting the atmosphere. It leaves half a cylinder of air available on either side at moderate pressure, to assist the next working stroke.

Productivity Measurement

IN general it is true to say that the importance and value of accurate statistical knowledge are still not realized throughout industry in this country. It was therefore, welcome news when some two years ago The Institution of Production Engineers and The Institute of Cost and Works Accountants appointed a strong Joint Committee to consider problems of productivity, a subject that can be discussed profitably only if accurate statistical information is available. This committee has recently issued a booklet, *Measurement of Productivity—Applications and Limitations*, which is well worth close study.

One of the first difficulties encountered by the committee was the lack of accurate information. To quote the report, they found :—

"(a) A lack of knowledge of the extent to which capital equipment in British industry is actually used. It is clear that machine utilization is low ; if it could be improved, a significant contribution to increased productivity would thereby be effected.

"(b) A lack of cost and other statistical information available at supervisory level. Much of the existing information is made ineffective through poor presentation and lack of understanding by the recipient.

"(c) A serious lack of agreed principles in the application of time-study methods. In many instances deficiencies in present time-study standards prevent their use for accurate comparison between departments or firms. Valuable additional data would become available if there were uniform methods of arriving at these standards."

It is obviously impossible to lay down any rule for the measurement of productivity that will be of general application to industry as a whole. The factors affecting productivity vary too widely, not only between one industry and another but even between firms engaged in comparatively similar activities.

Although the committee have not been able to advance definite recommendations concerning the methods for measuring productivity, they do make some extremely interesting suggestions about machine utilization and the presentation of information at supervisory level.

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THE NEW FORD CARS

A Detailed Description of the "Consul" and "Zephyr" Models

ENTIRELY new Ford designs constitute something of an event in automobile engineering. By using the same major components, including most of the body presswork, two models are provided. One has a four-cylinder engine for the economy market, and the other, with two additional cylinders and more luxurious specification, comes into a somewhat higher category.

The resulting manufacturing and stock-holding economies are obvious, and in the Ford case the most expensive tooling equipment covers both models. The fully automatic, electronically-controlled cylinder block transfer machine, can, for example, be employed without any stoppage or tool change on four- or six-cylinder blocks at will.

Although Dagenham-built, the designers were, of course, able to draw upon the resoures of the Detroit organization, and the main features constitute what may be termed a revolution in Ford practice. The name of Ford has in the past been associated with a number of major and minor design features, unique of their kind. At the time of their origination they were brilliantly adapted to the user and manufacturing conditions then obtaining.

Nearly all of these features have been discarded in the models now under consideration, which are, with a few exceptions, almost orthodox in design and construction. From the day of the first Model T, Ford cars have had transverse front springs with diagonal torque and radius stays, while the back axles were similarly sprung, with ball-jointed torque-tubes, stayed diagonally. A novel and ingenious coil-spring independent suspension is now used at the front, while the back axle has norma half-elliptic springs and Hotchkiss drive.

SPECIFICATION

ENGINE: Bore and stroke 3.125 *in* × 3 *in. Compression ratio* 6.8: 1. *Overhead valves.*

"Consul": four cylinders. Capacity 1,508 *c.c. Maximum b.h.p.* 47 *at* 4,400 *r.p.m., maximum torque* 74 *lb-ft. at* 2,400 *r.p.m., maximum b.m.e.p.* 121 *lb/sq. in.*

"Zephyr": six cylinders. Capacity 2,262 *c.c. Maximum b.h.p.* 68 *at* 4,000 *r.p.m., maximum torque* 112 *lb-ft. at* 2,000 *r.p.m., maximum b.m.e.p.* 122 *lb/sq. in.*

CLUTCH: Single disc, 8 *in diameter, hydraulic withdrawal.*

GEARBOX: Three speeds, baulked synchromesh on top and second, column control. Top 1: 1, *second* 1.642: 1, *first* 2.84: 1, *reverse* 3.86: 1.

REAR AXLE: Banjo, hypoid. "Consul": 4.625: 1. *"Zephyr":* 4.375: 1.

BRAKES: Girling 9 *in diameter,* 1¾ *in wide.* 2 *L.S. front.*

FRONT SUSPENSION: Independent, coil spring, 7.21 *in static deflection.*

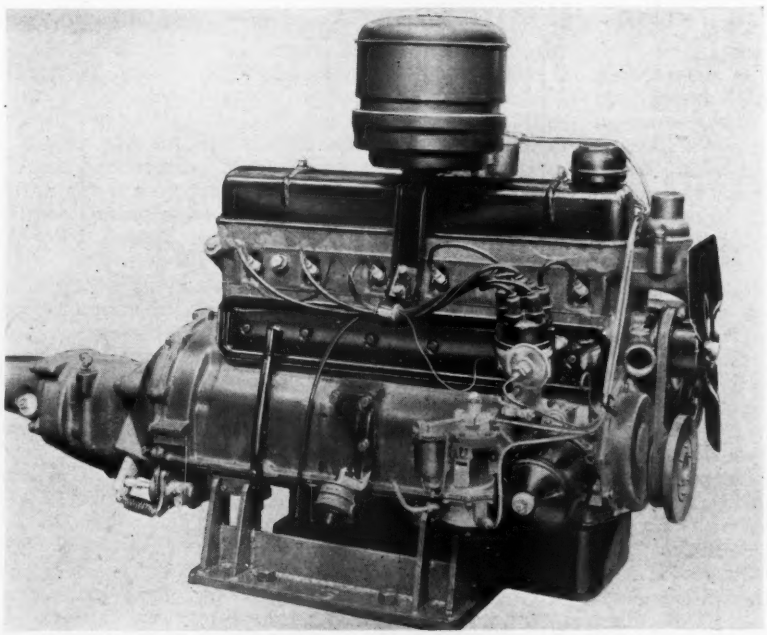
REAR SUSPENSION: Leaf springs, 5.625 *in static deflection.*

DIMENSIONS: Wheelbase, "Consul" 8 *ft* 4 *in, "Zephyr"* 8 *ft* 8 *in. Track, front* 4 *ft* 2 *in, rear* 4 *ft* 1 *in. Tyres, "Consul"* 5.90 *for* 13 *in rim, "Zephyr"* 6.40 *for* 13 *in rim. Kerb weight, "Consul" front* 1,306 *lb, rear* 1,044 *lb. "Zephyr" front* 1,479 *lb, rear* 1,063 *lb.*

For many years past the side-valve engine, with large-diameter split guides and upset valve stem tips, has been standard Ford practice. In the new overhead-valve engines the orthodox fixed, small-diameter guides and split cotters are fitted. Another move towards more conventional design is the abandonment of the integral connecting rod and stud construction in favour of a drilled rod and separate bolts.

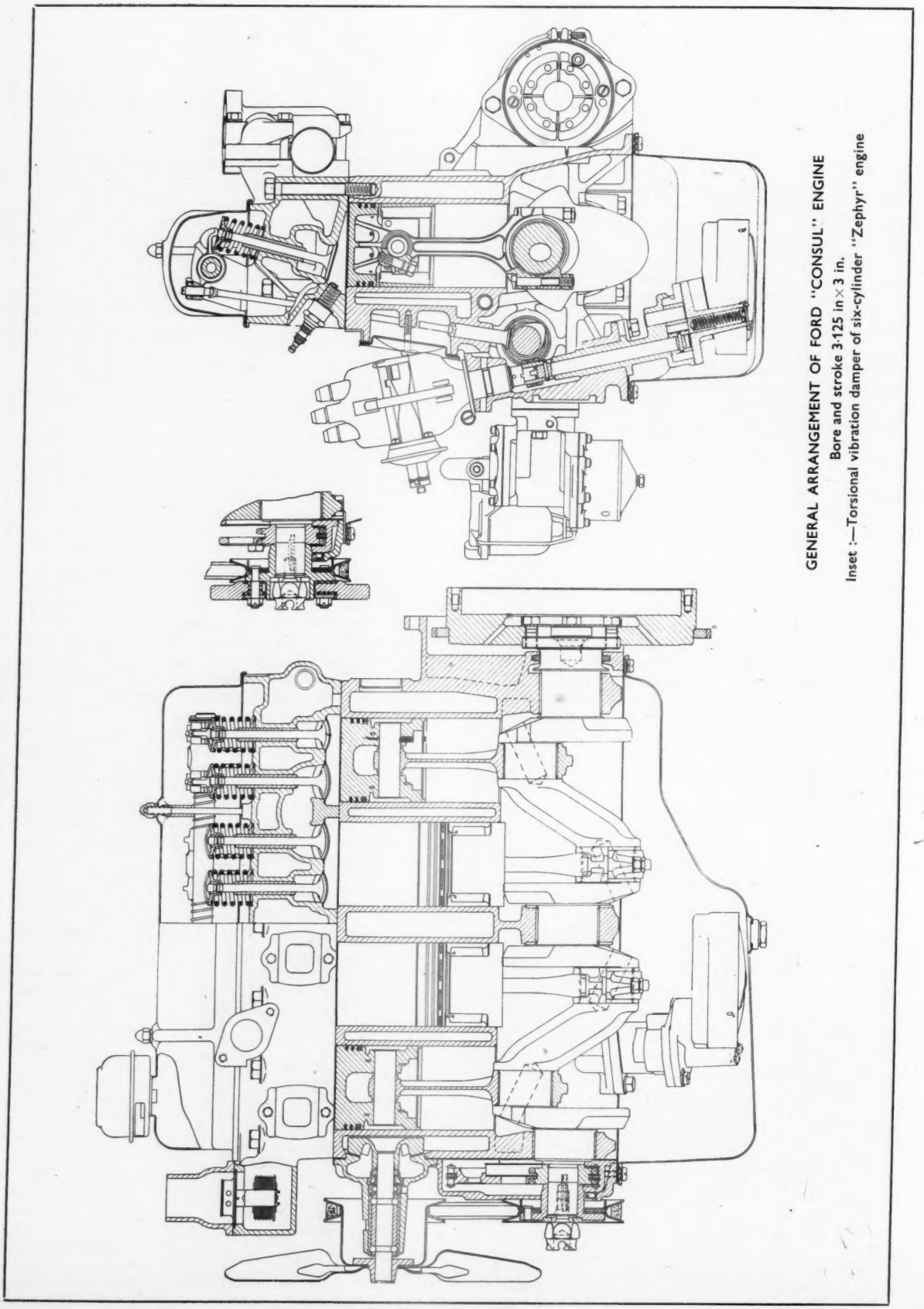
Until recently the Ford back axle was built up of a centre ring with integral pinion bearing housings and two bolted-on side trumpets carrying the differential bearings. This construction has always seemed to be lighter, stiffer and slightly less costly than a banjo axle. It has nevertheless been given up in favour of the more orthodox construction. This is all the more interesting since the Nuffield Organization, on all its products except the Riley, some time ago dropped the banjo axle in favour of a design similar to that just abandoned by Ford.

Among the Ford special features that are continued, are the cast crankshafts. Incidentally, the Ford Motor Company, over forty years ago, were the originators of the detachable cylinder head. This arrangement has both in foundry and machine shop, contributed more than any other design point to the ease of manufacture of the modern automobile engine.



Ford "Zephyr" six-cylinder engine

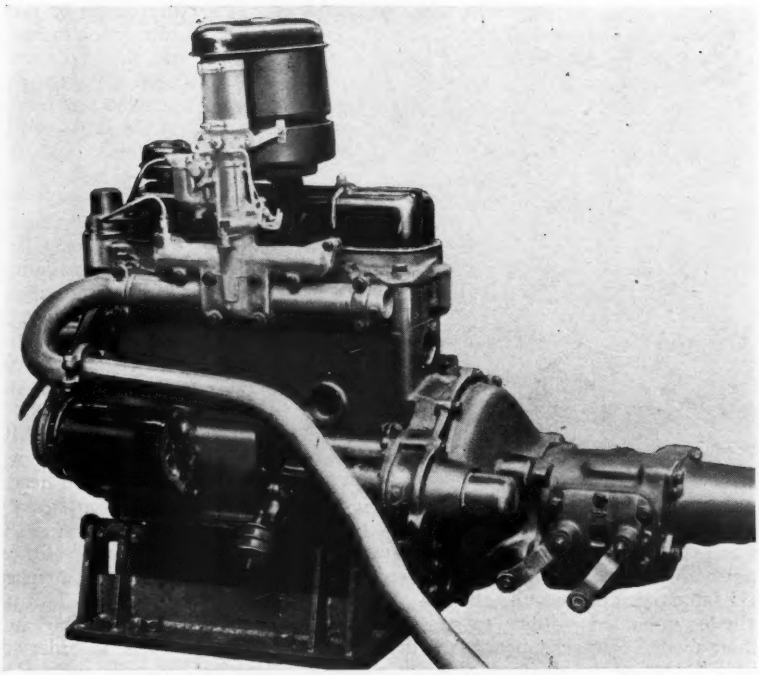
In general appearance the new "Consul" and "Zephyr" cars may be described as following the lines of the larger American Ford models. Integral body and chassis construction and detail weight-reduction



GENERAL ARRANGEMENT OF FORD "CONSUL" ENGINE
Bore and stroke 3·125 in × 3 in.
Inset :—Torsional vibration damper of six-cylinder "Zephyr" engine

features such as the use of tyres of specially small diameter on 13 in rims, have kept the kerb weights down to 21 and 22·5 cwt respectively. The bodywork does, nevertheless, provide adequate legroom and a clear width between door pillars of nearly 50 in.

Overall height has been kept down to 5 ft 0·75 in and the rear floor height is just under 1 ft. In spite of the use of a hypoid axle, a tunnel 4½ in high is necessary. Both engines have the same bore of 3·125 in and the notably short stroke of 3 in, the capacities being 1,508 and 2,262 c.c. respectively.



Ford "Consul" four-cylinder engine

Though three speeds only are provided, the smaller engined "Consul" model gives an excellent all-round performance. A maximum speed of 73 m.p.h. is possible, with the engine turning at about 5,000 r.p.m. on an axle ratio of 4·625 to 1 and tyres having a rolling radius of about 11½ in. The short stroke overhead-valve engine does not seem to mind high speeds, gives 47 b.h.p. at 4,400 r.p.m. and a maximum torque of 74 lb-ft. at 2,400 r.p.m., corresponding to a b.m.e.p. of 121 lb/sq. in.

Over 40 miles per hour can easily be attained on second gear without undue fuss and the general behaviour of the engine is such that the lack of the high-geared third speed provided by the customary four-speed gearbox is not seriously felt.

While most of the design is surprisingly orthodox, for the Ford Motor Company, the front suspension is unique in that coil springs act direct on the swivels. They are guided at their upper ends by the telescopic shock absorbers and at the bottom by lateral links stayed in a fore-and-aft direction by the arms of the anti-rolling bar.

Engines

Bore and stroke, at 3·125 in and 3 in respectively, are common to both "Consul" four- and "Zephyr" six-cylinder engines. The combustion chambers, pistons, valves, etc. are also common and a compression ratio of 6·8: 1 is used. In proportion to its capacity of 1,508 c.c., the "Consul" engine is given rather more manifold breathing capacity than the 2,262 c.c. "Zephyr". As a result the specific horse-power of the "Consul" is somewhat greater, as can be seen from the fact that it peaks at 47 b.h.p. at 4,400 r.p.m., whereas the 50 per cent. larger six-cylinder engine gives only 68 b.h.p. at 4,000 r.p.m.

At lower speeds, however, the slight advantage of the six cylinders is apparent, maximum torque at 112 lb-ft. at 2,000 r.p.m. comparing with 74 lb-ft. at 2,400 r.p.m., giving maximum b.m.e.p. figures of 122 and 121 lb/sq. in respectively.

Push-rod operated valves lie at an angle of 14 deg in combustion chambers having wedge-shaped boundaries giving a volume-controlled combustion. The general layout is robust and generous, all bores are, for example, completely water-jacketed, the space being nowhere less than 0·312 in. The cylinder blocks are carried well below crankshaft centreline for the first time in the history of the Ford Motor Company.

In view of the high revolution rates attained, special attention has been given to camshaft and valve spring design. Interest also attaches to the novel exhaust manifold, which consists of a steel tube held, without interposition of gaskets, in broached recesses in the cylinder head.

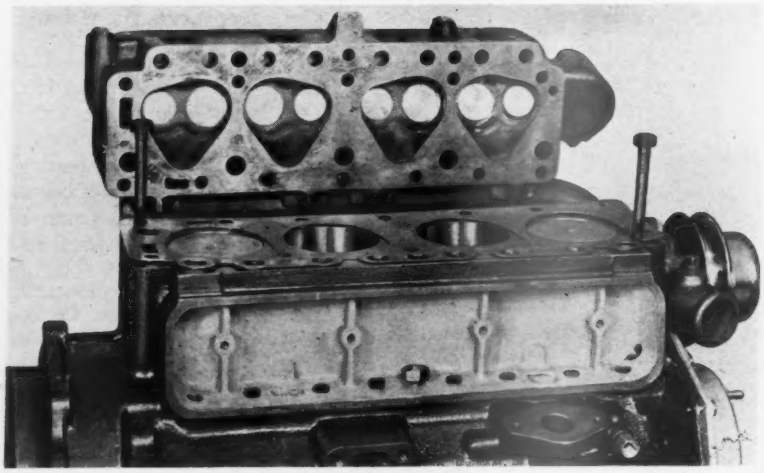
Cylinder block

The four- and six-cylinder blocks are identical except for the extra bearing and pair of bores. The oil pump is towards the front, between No. 1 and 2 cylinders in each case. Crankshaft end location is provided by white-metalled shims on either side of the last bearing but one, that is, the centre main in the four-cylinder and No. 3 bearing in the six-cylinder block.

The sump joint face is 2 in below crankshaft centreline and the cast-iron caps are registered in the recessed block and secured by $\frac{7}{16}$ in setscrews. Rear closure is by the usual filling-piece, grooved to take wedge-shaped half-packings of braided cotton treated with graphite and impregnated with wax. Following American practice, the flange of the pressed steel sump is corrugated. Two of the ribs run right round past the setscrews, while the centre pair are faded out at each setscrew seating. A cork gasket $\frac{1}{16}$ in thick is fitted.

While on the subject of joints, attention may be drawn to the cylinder head gasket, which is a single sheet of moderately stout bright steel with a single semi-circular rib raised in it, running fairly close to the edge of each hole. Both combustion spaces and water passages are dealt with in the same way and the gasket is dipped in synthetic varnish. The gauge of the material is such that there is no possibility of appreciably flattening out the corrugations. While this scheme presumably necessitates a high-class finish of the joint faces, the gasket is ideally simple and low in cost. Being moreover of considerable thickness without any asbestos, the conductivity should be quite good, in spite of its being made of ferrous material. Overheating of the edges would therefore seem to be unlikely.

On account of the remarkably short stroke the cylinder block is notably squat and broad in cross-section, the lower jacket face, for example, is only 2·625 in above crankshaft centre line,



Cylinder block and combustion chambers

while on the other side of the block the oil gallery combines with the web supporting the integral tappet guides and the enclosure of the low-mounted camshaft to form a girder about 3·5 in in horizontal width. This makes a considerable contribution to the lateral stiffness of a casting which is, in any case, nearly 9 in wide over the crankpit walls.

Unlike many English designs, there has been no attempt to arrange for through-milling or broaching of the side wall faces by eliminating timing case and flywheel housing projections and incorporating sandwich plates and the like. The integral flywheel housing is extended to the near-side to take the facing for the starter motor, while the flange to which the timing case is attached prevents pass-milling of the tappet cover joint face. Many extra parts are thus eliminated and a stiffer and more oil-tight job results. Machining costs are not affected since a special transfer machine has been developed on which each machined face is dealt with to its own requirements by appropriate tools.

Crankshaft and piston assembly

Cast in a special type of steel developed by the Ford Motor Company, the shafts of both four- and six-cylinder engines are counterweighted on each crank-web and have bearings between each pair of cylinders. All main bearings are 2·25 in nominal diameter and have thin-wall liners with effective lengths of 1 in for the front, 1·25 in for the intermediate and 1·5 in for the rear bearings. All crankpins are 1·937 in nominal diameter and the steel-backed big-end shells have an effective length of 1·125 in.

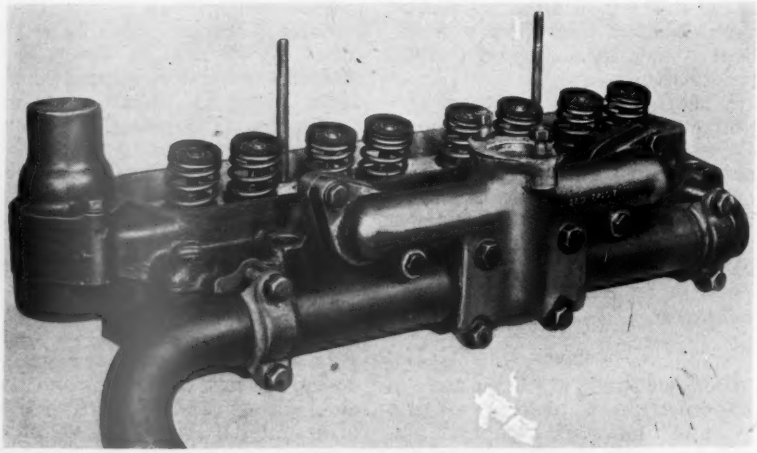
The crank webs are 0·75 in thick, except the flywheel end web, which is 0·875 in. Lightening holes are drilled diagonally in each crankpin and as a result, the oil drillings have to be offset to miss them. The offset happens to put the holes in a "leading" position, although a "trailing" location is more usual. The alloy steel connecting rods are 5·312 in centres and have $\frac{5}{16}$ in bolts secured by pressed steel locknuts of the Palnut type. The gudgeon pins are clamped in the rods and are $\frac{13}{16}$ in diameter.

The Autothermic aluminium alloy pistons have steel struts and solid skirts with a slipper cross-section at the gudgeon pins. One slotted scraper ring is fitted and two compression rings with stepped upper faces to give a slight scraper action. An important detail is the off-setting of the crankshaft and gudgeon pin plane by 0·0625 in in a trailing direction from the plane of the cylinder bores. This gives a better control of the piston change-over at top dead centre and reduces any tendency to slap. It also brings the centre of thrust pressure due to connecting rod angularity further down the skirt than would otherwise be the case. Bearing pressures are thus more equalized as between top and bottom of the skirt, and the useful life of the pistons should be extended.

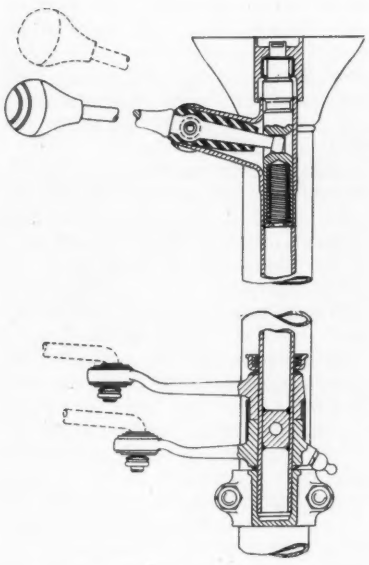
Cylinder head and valve gear

The combustion chambers are roughly triangular in plan, the "squish" turbulence being provided by the fore-and-aft edges and not, as is more usually, by an edge opposite the valves. Compression ratio is 6·8 : 1 and the valves, and the top of the combustion chamber, are inclined at 14 deg. Having normal 45 deg. seats the inlet valves are of SAE-8645 steel and have port diameters of 1·2 in. The exhaust valves have heads of DTD.49B steel welded to stems of En.18B, the port diameter being 1·0 in in this case. The lift is 0·35 in for both inlet and exhaust.

Interest attaches to the valve springs, the lower $2\frac{3}{4}$ turns being pitched to give only 0·01 in clearance with valve open, in order to damp out, by progressive contact, any incipient "surge". Somewhat unusual is the fact that the valve rockers are of pearlitic malleable iron, the bores being phosphate coated and working on an induction-hardened carbon steel rocker shaft. Rounded foot tappets work on cams ground to a taper of 8 min to ensure rotation. The cast nickel-chrome-molybdenum steel camshaft has cams of multi-sine-wave form somewhat similar to those developed by the Austin Motor Co. The shaft runs in three (or four) steel-backed babbitt bearings and is driven by a non-adjustable duplex roller chain of $\frac{3}{8}$ in pitch. All bearings are 1·765 in in diameter by 0·72 in long and all have forced lubrication with an intermittent jet from the front bearing to the timing chain.



Inlet and exhaust manifolds of "Consul" engine



Assembly of steering column gear control

Lubrication

There is the customary skew gear drive from the camshaft for the gear-type oil pump, with the usual dog coupling to the ignition distributor. A gauze suction strainer is fitted, and the pressure relief valve is of the plunger type, mounted in the pump cover. Oil is delivered to a full-flow filter of the bakelized corrugated paper type, located on the side of the crankcase, a by-pass relief valve is, of course, fitted to the filter. The main drilled oil gallery feeds the main and camshaft bearings with pipes to the rocker shaft. The connecting rods are not drilled for oil spray.

Manifolds and carburettor

On both four and six-cylinder models the exhaust manifold consists of a 1·75 in diameter steel pipe, with four or six ports cut in it. It is clamped in semi-circular recesses by pressed steel clamps and by a central depending lug cast integral with the aluminium alloy induction manifold and serving as a conduction hot-spot. No gasket is used and the end of the pipe bends downwards to take the welded flange that couples it to the main exhaust pipe.

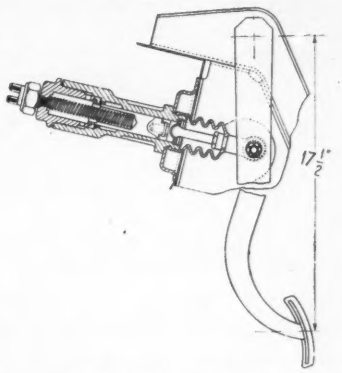
The "Consul" has a normal two-branched manifold while on the "Zephyr" the three-branched manifold has spiral ribs cast in its two lateral arms to assist in distribution. Zenith downdraught carburettors are fitted in both cases, a dry mesh type intake silencer being standard, with a combined oil bath cleaner and silencer as an optional extra. Fuel feed from the 9 gallon rear tank is by an A.C. diaphragm pump driven from the camshaft and combined with a double-acting vacuum pump to supplement manifold depression in operating the screenwipers.

Cooling

The tube and fin type radiator has a pressure cap set to 4 lb/sq in, both blocks are 2·34 in deep and 15 in high, but the width for the "Consul" is 13·86 in while the "Zephyr" has a block 19·98 in wide. The two-bladed "Consul" fan, driven in triangle with the dynamo by V-belt, is 12·8 in diameter, that on the "Zephyr" being 14·12 in. Water is circulated by centrifugal impeller on the fan spindle, with integral ball bearing and shaft and a carbon ring gland. There is the usual bellows-type thermostat with by-pass passage.

Electrical equipment

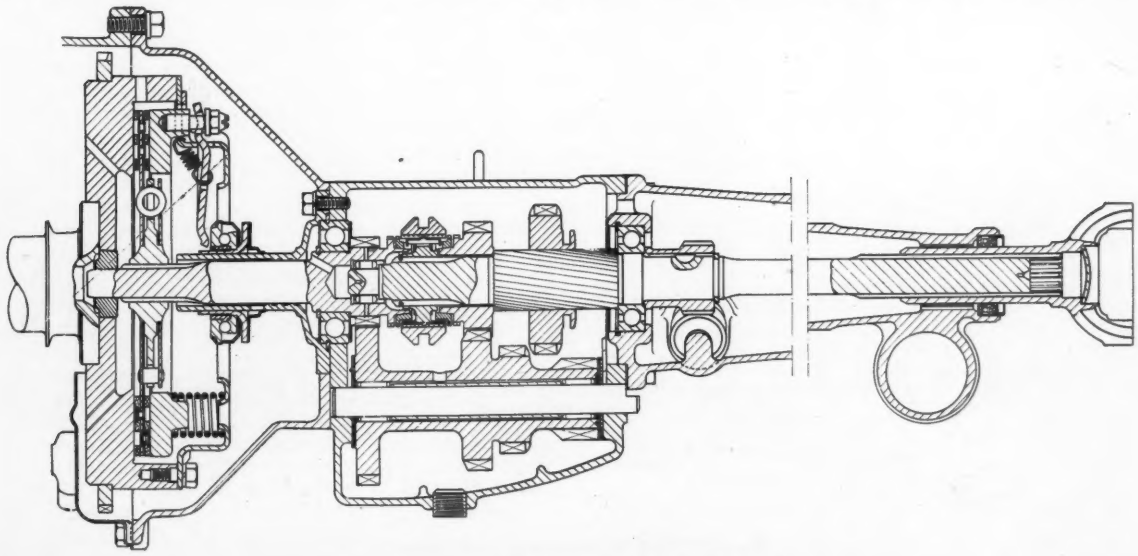
A 12-volt system is fitted to both cars, the shunt-wound dynamo having a maximum charge rate of 22 amp, voltage regulated, the drive ratio being 1 : 1·46. The starter has Bendix drive and a pull-type mechanical switch. Full vacuum control by integral diaphragm is applied to the distributor, the full throttle maximum advance being 16 deg at 3,500 r.p.m. and 10·75 in mercury manifold depression on the "Zephyr". The figures for the "Consul" are 14 deg at 4,000 r.p.m. and 9·5 in mercury. Champion NA8 14 mm. plugs are fitted as standard.



Mounting of clutch pedal and master cylinder

Clutch

The 9·625 in diameter cast iron flywheel has a shrunk-on starter ring and carries an 8 in dry disc clutch of Ford design and normal construction. The inside diameter of the lining is 5·25 in and a cushioned centre is used.



General arrangement of three-speed and reverse gearbox

The withdrawal bearing is of the permanently lubricated ball thrust washer type, sliding on the nose of the gearbox driving shaft bearing cover. Operation is by a pressed steel fork, swinging on a spherical pivot, the opposite end being thrust forward by a small hydraulic cylinder with adjustable push rod. Connection is made by flexible hose to one of a pair of Girling-type master cylinders mounted on the scuttle bulkhead and worked by pendant pedals for clutch and brake.

This arrangement isolates clutch engagement entirely from power unit vibrations or movements of any kind and has the additional advantage that no pedal slots are required. A return spring keeps the withdrawal fork normally out of engagement, so that the thrust bearing can revolve as one piece. Adjustment for wear is therefore required. It will be remembered that in a Patent taken out by the Rover Company, hydraulic operation was combined with a ball withdrawal bearing lubricated from the gearbox and suitable for continuous engagement. In this case no adjustment is necessary, the action being similar to that of the "Hydrastatic" brake.

Gearbox

The gearbox gives three speeds, the ratios being :— top 1 : 1, second 1·642 : 1, first 2·84 : 1, reverse 3·86 : 1. All gears are 9 D.P. and are helical, this necessitating helical splines for the sliding first and reverse wheel. The centres are 2·83 in and face widths about 0·562 in.

The baulked synchromesh differs slightly from the normal. Three deep keyways in the synchronizer hub carry three hollow pressed steel keys, the outer faces of which, have raised dimples engaging the usual internal groove in the striking ring. Radial pressure for "first intention" synchronizing is given by two spring rings bearing on the insides of the keys. These fit in clearance recesses in the bronze friction rings and contact the bottoms of the recesses to transmit synchronizing pressure.

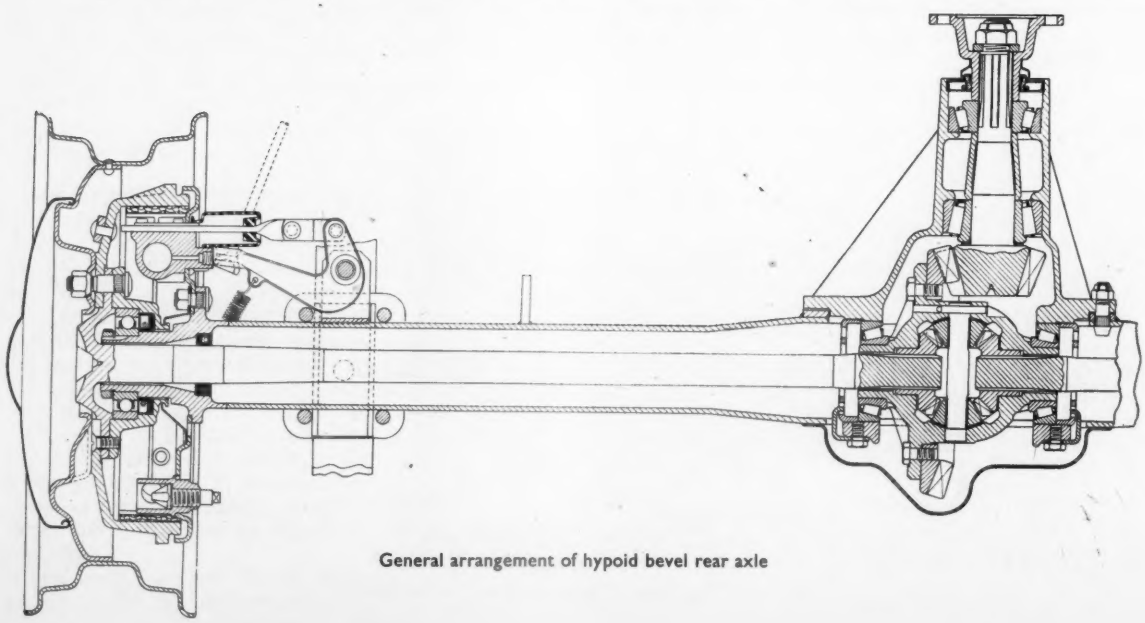
A steel-backed lead-bronze bush is used for the second speed driven wheel while the one-piece layshaft runs on needle roller bearings. The driven shaft is carried by a ball bearing secured by circlip to the long gearbox extension casting, which spigots in the box on a diameter large enough to clear the first speed wheel. Assembly of the whole driven shaft can therefore be made before insertion in the box. Fifteen inches behind this bearing there is a steel-backed bush supporting the front jaw of the Hardy Spicer propeller shaft which slides on the driven shaft. The oil return holes in the extension are so positioned that, even with a low level in the main box, the bottom of the support bush is fully immersed.

Under the end of the gearbox extension there is an eccentric bonded-rubber bush that forms the rear support of the power unit. The thinner segment of the rubber is actually cored away, the effect being to give considerable flexibility in a vertical direction but stiffer location in a fore-and-aft sense. The front support is by two metal-interleaved bonded rubber sandwich pads attached to the cylinder block at such an angle as to bring the oscillation centre well up in the region of the mass-centre line. The striking gear follows lines that seem to be practically standard for three-speed gearboxes, short internal cranks with integral sectors for the plunger notches carrying glut pieces engaging the striking collars.

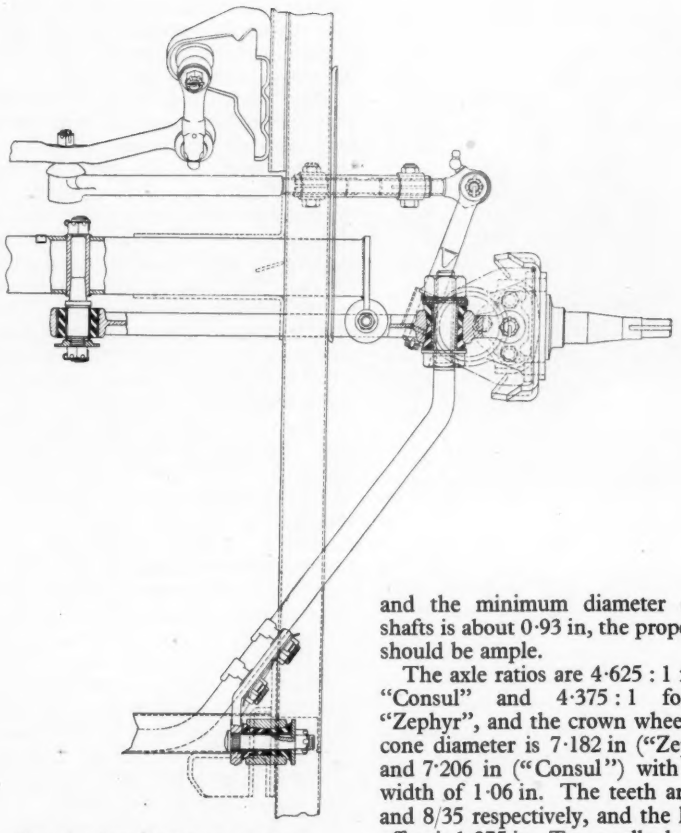
The external levers are, as usual, about three times as long as the internal cranks and carry rubber-cushioned bushes in which fit the bent-over ends of the coupling rods. Selection at the lower end of the column control is by a tube sliding through the two levers and stiffened locally by an internal plug brazed in position to take a cross-pin engaging jaw slots cut in the lever bosses. A thin steel cover tube keeps the levers in position and excludes dirt. The change-speed lever pivots in a tubular lateral extension of the column control tube, its ball-shaped inner end fitting in a spindle carried at its upper end in a bush on the steering column. A spring is fitted between the spindle and the column tube, which, by pressing the tube downwards, brings the lever normally into the lower postion for top and second speed. This is a reversal of customary practice, and there does not appear to be anything to recommend the arrangement. A vital point is the surrounding of the pin joint on the change-speed lever by a snugly fitting rubber sleeve.. Not only does this stop all rattle but it cushions the whole mechanism. It makes a definite contribution to a good example of column-control which certainly functions excellently.

Back axle and suspension

The banjo casing has a welded-on rear cover and a fillet-welded stiffening



General arrangement of hypoid bevel rear axle



Plan of independent front suspension

ring at the nose seating. The tubes are 2·5 in diameter by about 0·156 in thick and the hub ends are flash-welded on. The usual channel-section seatings are welded on to take the springs, which are 1·75 in wide and underslung, and have seven leaves. The spring centres are 42 in and the static deflection 5·625 in under a design load of 620 lb for the "Consul". The two top leaves are 0·225 in thick and the remaining five are 0·185 in. On the "Zephyr", the spring details are identical, the extra weight of the six-cylinder engine being taken entirely on the front suspension. Double-cylinder lever-type shock absorbers are fitted and the springs have Silentbloc type bushes at the front with Harris-type bushes for the rear shackles. An important detail is the fitting of rubber pads at the tips of the tapered spring leaves.

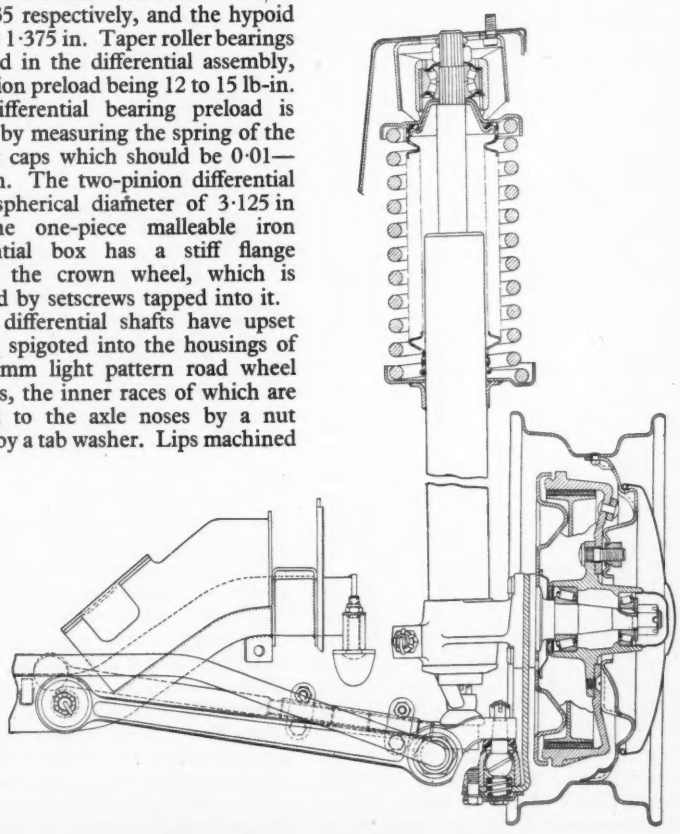
A superficial examination of the differential and shaft assembly might suggest that these parts are on the light side for a car, the "Zephyr", with 68 h.p. available and a kerb weight of about 2,600 lb with full equipment. The "Consul" has 47 h.p. and a weight of 2,350 lb. As, however, the wheels are of specially small size, having a rolling radius of 11½ in and the minimum diameter of the shafts is about 0·93 in, the proportions should be ample.

The axle ratios are 4·625 : 1 for the "Consul" and 4·375 : 1 for the "Zephyr", and the crown wheel pitch cone diameter is 7·182 in ("Zephyr") and 7·206 in ("Consul") with a face width of 1·06 in. The teeth are 8/37 and 8/35 respectively, and the hypoid offset is 1·375 in. Taper roller bearings are used in the differential assembly, the pinion preload being 12 to 15 lb-in. The differential bearing preload is judged by measuring the spring of the bearing caps which should be 0·01—0·012 in. The two-pinion differential has a spherical diameter of 3·125 in and the one-piece malleable iron differential box has a stiff flange behind the crown wheel, which is attached by setscrews tapped into it.

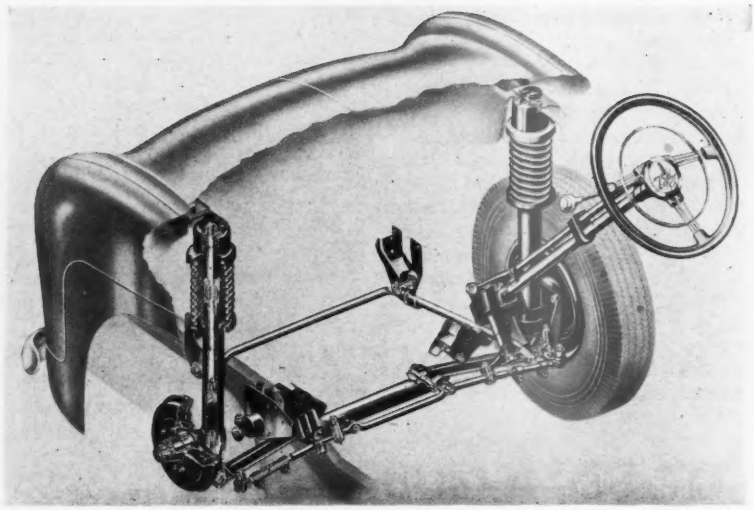
The differential shafts have upset flanges, spigoted into the housings of the 40 mm light pattern road wheel bearings, the inner races of which are secured to the axle noses by a nut locked by a tab washer. Lips machined on the inner ends of the housings throw any oil that passes the oil seals into pressed steel catches secured by the backplate bolts. Spigoted on to registers on the outside of the differential shaft flanges, the cast-iron brake drums are secured by the through-going wheel bolts, the serrated shanks of which take a bearing in the flanges of the bearing housings. The usual pair of cone-headed setscrews serve to keep the parts in position when the road wheel is removed.

Brakes

With drums 9 in in diameter and shoe linings 1·75 in wide, the front brakes are Girling two-leading shoe, with sliding shoes, and snail cam adjusters. Single leading shoe brakes are employed at the rear, with sliding shoes and cone adjusters. The hand-brake is worked by wedge and roller mechanism housed in a recess on the outside of each wheel cylinder. In place of the customary swivel-tree, each brake expander has its own bell-crank pivoted on a pin carried by a pressed bracket welded to the axle arm just over the spring seating. From these bell-cranks, diagonal rods run to an equalizer from which a cable connects to the handbrake lever. The total lining area for the four wheels is



Elevation of front suspension and arrangement of hub



Layout of front suspension and steering assembly

121 sq in and the nominal thickness 0·187 in. Footbrake operation is by Girling plunger-type master cylinder secured to the scuttle alongside the similar cylinder for the clutch operation, both being operated by hanging pedals.

Front suspension and steering

The highly original independent front suspension is an excellent example of the way in which present-day body design influences the practical value of a suspension layout. Twenty years ago when the frame was the foundation of the car structure and the bonnet and wings were separate entities, the Ford design would doubtless have been criticised. With a swivel assembly standing about two feet above wheel centre-line, it would have been regarded as a clumsy scheme, distributed all over the place, spoiling the look of the car and requiring heavy upstanding brackets to take the load so high in the vertical plane.

Nowadays, however, when the inside of the wing is hidden and forms part of a rectangular, concealed box enclosing the engine, the new Ford suspension is completely out of sight Further the sides of the deep box that takes the place of the wing valances, are quite capable, with a little local stiffening, of dealing with the vertical thrust of the suspension coil springs. Again, by spacing the upper and lower attachments some two feet apart, the torsional stresses formerly set up by braking strains in the suspension cross member of an orthodox wishbone suspension are reduced to rearward and forward forces of quite small magnitude, and a great deal of weight can be saved at the lower end. The absence of attachment points for the inner ends of upper wishbone links may also make for a little more space around the engine. The layout thus becomes if not actually accessible, at least rather less inaccessible than the engine in the average modern car.

There is light tubular cross member in the sub-frame that forms an integral part of the combined body and chassis. This carries two pins on which are mounted, on bonded rubber bushes of short length and considerable thickness, the inner ends of laterally extending radius arms. These are some 15 in long, ball-jointed at their outer ends and are inside pressings that form part of the front swivel assembly. Fore-and-aft location of these arms is provided by the anti-rolling bar, the rearwardly-extending portions of which pass through the radius arms and are fitted with coned rubber bushes at this point. Attachment of the anti-rolling bar to the frame takes the novel form of two stampings, split, clamped and dowelled to the bar. They carry, on overhung pins, two conical rubber bushes that seat in stampings bolted to the sub-frame side members.

The working movement of this assembly involves geometrical anomalies which are quite satisfactorily dealt with by the accommodating rubber bushes. In particular, the rise and fall of a radius rod is accompanied by a considerable twisting by the radius-rod action of the anti-rolling bar arm. This is taken up by the specially-proportioned short bush at the inner end. The ends of the anti-rolling bar also open and shut as the radius arms describe their arcs. This movement is taken up by bending of the bar and by cross-deflection of all the conical bushes. As all the arms are, however, about 15 in radius, no harm is done. The scheme is, nevertheless, one that is based on the fact that rubber bushes have now been fully developed.

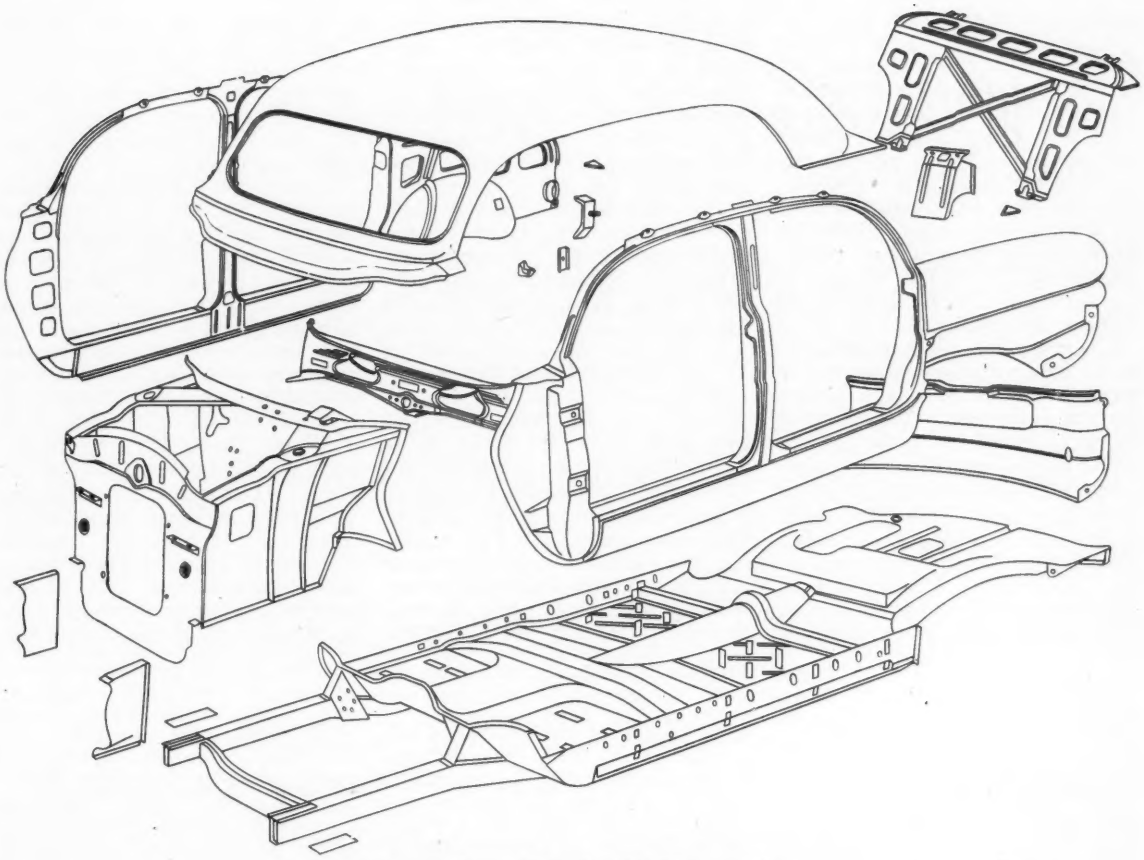
The upstanding portion of the swivel assembly takes the form of a long telescopic shock absorber with a working bore of 1·375 in and a spindle diameter of 1 in. The absorber is much longer than its working stroke requires, about 6 in of the travel being stopped by an internal distance tube so that there is always plenty of spindle inside the absorber. The spindle guide at the top has a spherical mounting in the body tube, ensuring a full bearing under lateral load without bell-mouthing of the bush.

Above this spherical bearing, suitable seals are fitted which should retain the fluid for the life of the vehicle, although provision is made for topping up if this should become necessary. Suitable guards are also fitted to prevent damage to the central spindle. It will be noted that there are only two lubrication points on this independent system, these being the ball joints on the radius arms.

The coil spring surrounds the absorber and has $9\frac{3}{4}$ active coils of a mean diameter of 4·125 in and is made of material 0·472 in diameter. The rate is 105 lb/in and the deflection about 7 in under normal static load. The spring bears on pressings carried by the spindle and the absorber body tube, these having helical seatings so that grinding of the end coils of the springs is eliminated. Steering movement is taken by an opposed taper roller bearing on the absorber spindle. Its outer race is housed in a pressing bonded to a rubber sleeve in a pressing bolted to a stiffened part of the combined engine box and wing valance. Fore and aft loads are taken by a pressing running diagonally to the scuttle bulkhead. A rubber pad limits the bump travel to about 3·5 in but rebound is permitted to about 4·5 in below. It is dealt with by the absorber alone, without check stops.

The absorber has a disc suction valve in the foot and a poppet discharge, the valves in the piston being of poppet type, one being carried within the other. Tested at 104 cycles/min of a 6 in stroke, the standard valve setting gives a fairly flat curve at about 200 lb for rebound and 100 lb for bump. Low-speed leak settings are about 20 lb in each direction.

Since the radius arm at the bottom has a finite length while the sliding top attachment is equivalent to a link of infinite length, there is appreciable variation of track as the wheels rise and fall. For 2 in above normal the track spreads about 0·375 in, beginning to close in again for further upward



Components of integral chassis and body construction

movement. A 2 in downward movement gives rather more than 1 in reduction of track, while at 4 in rebound, the track reduction is about 2½ in. Though academically undesirable, experience with cars embodying much worse figures than this shows that the effect on road handling and tyre wear is negligible. The steering of the new Ford is certainly beyond reproach. It is also true that there is rather more change in camber angle than on some wishbone suspensions, the camber increasing by 2 deg for 4 in rebound. The figure for the more normal movement of 2 in below normal is, however, only 0·8 deg. There is in any case not the slightest sign of gyroscopic "kick" in the vehicle.

The tolerances of the settings for kerb load are :—

Toe in ..	0·062in to 0·125in ..	adjustable
Castor ..	+½ deg to −½ deg	adjustable
Camber ..	1½ deg to zero	non-adjust.
K.P.I. ..	2¾ deg to 4 deg	non-adjust.
Toe out. 20 deg turn outer wheel ..	4½ deg to 5½ deg ..	adjustable

An outstanding characteristic in the handling of the car is the lightness of the steering under low speed conditions and also when cornering. For this, the wide spacing of the king-pin bearings is doubtless partly responsible. It is about 30 in as compared with say 3 in for an ordinary steering swivel, while the top bearing is a roller bearing, taking end thrust also. Lateral load, due to wheel offset or centrifugal force can therefore cause no appreciable increase in the friction of the system. Steering linkage is by ball-jointed rods about 14·5 in long paralleling the radius arms and connected to a centre rod carried at one end by the steering drop arm and at the other by a slave link. Threaded steel bushes are employed at these points. The peg and worm steering gear has a ratio of 13·6 : 1. The steering is quick and light, but a turning circle less than 40 ft 6 in might have been anticipated on a car with a wheelbase of only 8 ft 4 in and with such unusually small wheels.

Body and chassis construction

Taking the floor sheet as a basis, this is made up of two pressings welded together in front of the upsweep which clear the back axle. The front pressing includes the foot ramp and an inverted trough cross-indentation under the front seat. This, by the insertion from below of a trough pressing, is converted into a box cross member passing over the power unit rear suspension point, which is attached to a small cross-pressing passing underneath and secured by bolts. Just behind the rear seat footplates, there is a sharply upswept inverted trough member passing over the tunnel and spot-welded to the floor by its flanges. To the upturned flanges on the sides of the front pressing two deep vertical sheets are spotwelded and serve as closing plates for the body sills, which are about 6 in deep and come both below and above the floor.

Two inverted trough pressings, converted to box-section by top closure plates, pass under the floor, where their rear ends are attached to the power unit attachment cross member. They are also welded to the front seat floor plate and, through the intermediary of box gusset brackets, to the toe ramps, while a box-section cross member joins the front ends. The forwardly projecting sub-frame thus formed, takes the engine front supports and the lower attachments of the front suspension linkage and is greatly stiffened by the welding to it of the scuttle pressing, two full-depth side plates, and a front cross-bulkhead

pierced to take the radiator block.

Channel-section brackets welded to the outer faces of the side plates reinforce them at the attachment point of the front suspension shock absorber and load-carrying spindles. Lateral stiffness is increased by diagonal pressings picking up the forwardly-projecting edge of the scuttle pressing. At some loss of engine accessibility the front end of the car is thus rendered enormously stiff without the use of material heavier than 20 gauge for main components.

The rear of the vehicle is also reinforced by trough side members, the flanges of which are welded to both floor pressings and which serve as attachment points for the rear springs.

The load-carrying structure is completed by the door side frames and the inside rear panels with their wheel arches and half-depth cover pressings which bridge the wheel boxes at about half wheel radius and greatly increase the stiffness of the inside rear panels. The opening behind the rear seat cushion is spanned by two crossed pressings and the assembly is completed by the addition of the roof. The external side panels at front and rear are, of course, attached by screws and take no appreciable share of the body stresses. The bluff styling of the rear end of the car, in conjunction with the forwardly-located rear seats, gives a luggage compartment of really useful proportions. The spare wheel is carried in it, located edgeways in a well forming part of the off-side of the floorplate pressing.

General dimensions

With a wheelbase of 8 ft. 4 in the "Consul" has an overall length of 13 ft 6·5 in, the corresponding figures for the "Zephyr" being 8 ft 8 in and 14 ft 3·5 in. The maximum width of both cars is the same at 5 ft 4 in, the front wheel track is 4 ft 2 in and the rear 4 ft 1 in. The ground clearance at the clutch pit and body sills is just over 8 in for both cars, the clearance at rear axle housing being 6·6 in for the "Consul" and 7·1 in for the "Zephyr". The overall height, laden, is 5 ft 0·75 in for both. The "Consul" has tyres 5·90 on 13 in rims and the "Zephyr" 6·40 on 13 in rims also. The inflation pressures are 28 and 24 lb respectively.

As previously mentioned, both cars are distinctly heavy on the front, as compared with the back wheels. The "Consul" kerb weights are 1,306 lb front and 1,044 lb rear, increased to about 1,476 lb front and 1,154 lb rear with two 10-stone passengers in the front seats. For the "Zephyr", with heater and radio fitted, the front kerb weight is 1,479 lb and the rear 1,063 lb. With two 10 stone passengers in the front seats, the front weight is about 1,639 lb and the rear 1,173 lb, say 58 per cent front and 42 per cent rear. With four passengers and 1 cwt of luggage, the front wheel weight is practically the same as the rear in the case of the "Consul". The "Zephyr" has, however, approximately 150 lb more weight on the front than on the rear.

AUTOIGNITION OF ENGINE FUELS

A PAPER by W. J. Levedahl and F. L. Howard entitled "An Apparatus for Studying Autoignition of Engine Fuels" (Research Paper 2200) has been published in the *Journal of Research of the National Bureau of Standards*, Vol. 46, No. 4, April, 1951. The full description is mainly of interest to scientists engaged on fundamental combustion research and may be obtained from the Superintendent of Documents U.S. Government Printing Office, Washington 25, D.C.

Many investigators have recognized two major stages in the combustion of hydrocarbons. The first stage is characterized by a relatively slow reaction following an induction period whose duration depends on experimental conditions such as temperature, pressure, fuel-air ratio, presence of additives and the condition and area of the sufrace exposed to the mixture. Cold flames, blue luminescence, and relatively small changes in pressure are some of the physical phenomena associated with the first-stage reactions.

From a chemical standpoint the first isolable material from this reaction is peroxidic in character, and hydroperoxides have been identified. The exact mechanism of peroxide formation is unknown, but it is presumed to result first from the formation of short-lived free radicals from the hydrocarbon molecules, which in turn react with oxygen to form peroxides. Decomposition of these hydroperoxides gives rise to new oxygenated free radicals that act as carriers for the propagation of a chain reaction, some of whose intermediate products have been identified as aldehydes, ketones, and acids.

The progress of the first-stage reaction is governed by experimental conditions and by the molecular configuration of the fuel. It is unlikely that it ever reaches a condition of equilibrium in an engine, or even that a concept of equilibrium could be applied to the reaction. Possibly the first-stage reaction influences the second or "firing" stage of combustion by providing certain essential chain carriers. It may be that the second stage will not occur until the first stage has progressed to a certain point.

In a spark-ignition engine, normal combustion is initiated at the spark plug, and a flame front travels through the mixture from this point of origin, compressing the unburned mixture of fuel and air ahead of it. This unburned portion of the charge is frequently called the "end gas." The temperature of the end gas is increased due to adiabatic compression by the burning gases in and behind the flame front and by motion of the piston. This temperature rise, accompanied by increases of pressure and density, sometimes causes first-stage reactions to occur, often followed by a very rapid second-stage reaction, which may cause combustion knock if it takes place with sufficient rapidity. It is thought that investigation of the characteristics of these reactions may prove helpful in obtaining a better understanding of combustion knock.

In order to extend and augment work previously reported on compression ignition studies, a programme has been initiated to determine how the structure of the fuel affects the duration of preflame reactions and the pressure developed by these reactions. It is planned to use several six-carbon hydrocarbons as fuels. These compounds have a variety of structures; paraffins, naphthenes, olefins, and an aromatic being included. This paper describes the apparatus and the methods of operation, and presents the results of preliminary experiments in which *n*-heptane and *n*-hexane were used.

The paper includes a description of the apparatus and the experimental procedure, and sets out the results obtained with normal heptane and normal hexane. The conclusions arrived at in the report as a result of the work done are :—

1. *n*-heptane and *n*-hexane both evidence a pronounced two-stage autoignition reaction over a wide range of operating conditions.

2. When all other conditions are held constant, the pressure and crank angle at which the first-stage reaction begins in a steadily firing engine are nearly constant over a range of mixture compositions from critically lean to 1·2 times stoichiometric.

3. The duration of the first stage of the reaction decreases as the mixture is made richer up to about 1·2 stoichiometric, then remains nearly constant.

4. Considerable ionization occurs in the cylinder during firing, but none was observed during the first-stage reaction. The ion concentration is affected greatly by fuel-air ratio and is relatively unaffected by changing other operating variables.

5. When residual gases from either a first- or second-stage combustion are in the cylinder, the charge will burn earlier than when uncontaminated. (1964)

CAR ASSEMBLY

An Important Development at the Longbridge Works of The Austin Motor Company Ltd.

IN itself the new car assembly factory recently opened by The Austin Motor Company Ltd. is an outstanding achievement. If however, the full importance of this scheme is to be appreciated, the new factory must be considered as only part, although a very important one, of a comprehensive plan for the whole of the Austin manufacturing organization. This is a point that may easily be overlooked, since there are detailed developments in connection with the new factory itself which are completely novel and indeed unique.

As this new factory is an integral part of an overall planning conception for the whole of the Company's production functions, it is worth while briefly to discuss the basic principles employed. To-day, the rise in material costs is very much greater than the rise in labour costs with reference to pre-war prices. Because of this the Company's planning engineers attach great importance to two factors. These are the closest possible control of stocks and the reduction of contingency stocks to the lowest possible figure. The savings that will result from attention to these factors are much more important than the savings that will be made in labour costs. In fact, the direct labour costs for assembly will not be cut appreciably, except in so far as the wider use of mechanical equipment and the provision for bringing material in the right sequence to the various assembly stations will allow a given labour force to assemble a higher number of cars per day with less fatigue.

A further saving is in floor area. The new assembly lines and the stores feeding them occupy only about one-quarter of the floor area occupied by the lines they replace.

There will also be a direct and indirect effect upon product quality. The quality of the product as regards the final user will remain the same but it will be obtained with much less rectification than hitherto. In part this will be due to the much wider use of mechanical equipment during the actual assembly operations. Indirectly, the new assembly lines will probably have a reaction on earlier production processes, since there is no means of switching a faulty component or sub-assembly and replacing it by another one.

This new factory is completely integrated with the other production departments of The Austin Motor Company Ltd., but to obtain the maximum advantages from the planning conception, it is also necessary to have co-operation from the material suppliers. It is the deliberate policy of the Company to carry much smaller stocks than most similar organizations consider to be necessary. These stocks are, in fact, much lower than many Companies would consider safe minima.

This policy has two important advantages. In the first place only relatively small areas are needed for storage, and a higher proportion of the total floor area can be used for productive purposes. Secondly, the amount of capital locked up in stock is greatly reduced. Obviously, frequent deliveries of relatively small quantities from outside suppliers are necessary for the efficient working of this system. In the circumstances existing to-day, it is not always easy to obtain the necessary degree of co-operation, but several companies have already realized that the system is good for both the supplier and the user of the materials. In this connection it may be noted that the Company has arranged for a supply of Tasker trailers of standardized design on which unit loads can be transported from the supplier's factory to the Austin works. Special containers have been designed for carriage on these trailers. One company has found that the use of these trailers considerably eases their own problems of storage space, since the trailer, with its load of parts in a container, can be stored in the open until such time as delivery is required.

Supply to assembly

Power units rear axle and front suspension units, and bodies painted and trimmed, are supplied direct from the production departments to the appropriate stations on the assembly line. Other parts are issued in sets from the stores area in the new factory. Three assembly tracks have been installed in the new factory and there is provision for a fourth if it should be required. The three tracks are designed to give an output of 4,000 cars in 80 hours, that is, in a working week, since there is a two-shift system. Therefore, one complete car is turned out every 1·2 minutes. For a single assembly track the lowest cycle time is 2·4 minutes. This is for the assembly of A40 models on a track running at 6 ft. per minute. The track for A70 models runs at 4 ft. per minute and there is a correspondingly longer cycle time.

Three conveyor tracks have been installed for transferring main sub-assemblies from the production departments to the assembly tracks.

Front suspension units and rear axles are delivered on one conveyor system power units on the second and bodies on the third. These systems are so designed that at the correct point in the travel each load is automatically switched to the appropriate assembly track.

Without any doubt, one of the

Fig. 1. Engines on the special King marshalling conveyor from which they are released in sequence to the tunnel conveyor

Fig. 2. A body entering the tunnel from the loading station

Fig. 3. General view of the tunnel with three Dual-Duty conveyors

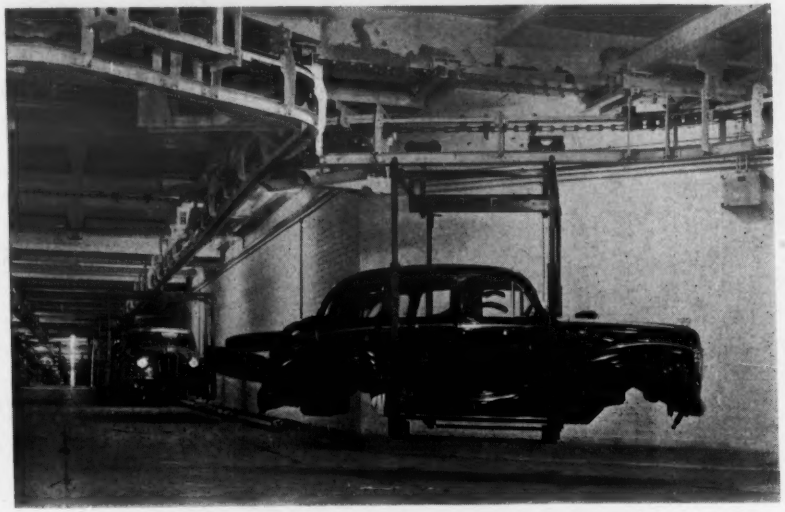


Fig. 4. A body entering the pit below the assembly tracks

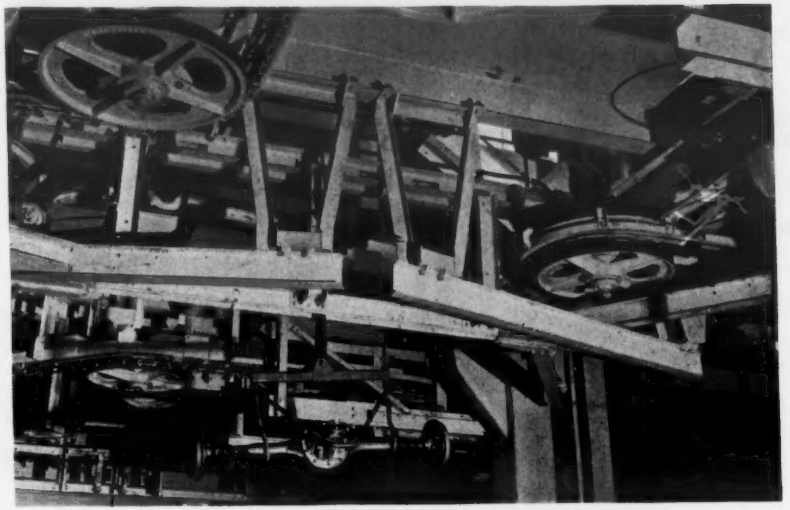


Fig. 5. A rear axle being automatically transferred to the elevator

most outstanding features of these conveyor systems is the manner in which the deliveries from them are correlated with the movements of the assembly tracks so that the various units reach their correct stations at exactly the same times as the chassis on which they are to be mounted. In order that this may be so, the first essential is that each conveyor must be loaded in conformity with a sequence laid down in the master production plan. For this planned sequential loading, the datum point is the start of body painting, which is 18½ hours before the finished vehicle will pass off the assembly track. The production of other units is planned in relation to this datum point so that they are loaded on to the appropriate conveyors at the time that will give complete correlation between the three conveyor systems and the assembly conveyors.

Because of the configuration of the site, the new assembly factory is at a considerably higher level than the production departments from which the various units are delivered. It has therefore, been found advisable to bring the various units through a tunnel into pits running some 20 ft. below the main floor level of the factory. Power unit assemblies and rear axle and front suspension units are completed in buildings some considerable distance from the assembly factory. For these, the conveyor sequence is: feeder conveyor from the production department to the marshalling area where they are automatically transferred to an indexing conveyor, thence by feeder conveyor through the tunnel for transfer to a second indexing conveyor, and thence to an elevating conveyor. The lay-out in one of the older buildings has been rearranged so that the building nearest the new assembly factory houses all the equipment for body paint and trim. Because of this it is possible to load the bodies direct on to the marshalling conveyor. From then on, the body conveyor sequence is the same as that for the other units. Incidentally, all of these conveyor systems have been designed and built by Geo. W. King Ltd., Hitchin, Herts.

Delivery of power units and rear axle and front suspension units to the marshalling area is effected by means of "Dual-Duty" conveyors. They differ from normal practice only in so far as they run at speeds considerably higher than are usually employed, and the spacing of the pusher dogs is wide, so that only a relatively small number of units is on the conveyor at any one time. On arriving at the marshalling area, the load is automatically transferred from the feeder conveyor to a special indexing conveyor. The conveyors in the marshalling area run at a lower speed than the feeder conveyors. This allows a certain buffer stock to be built up in the marshalling area. Actually, there is provision for 45 minutes supply of power units and 15 minutes supply of rear axle and front suspension units. These reserves are required because the final assembly factory and the departments producing the units do not work to the same hours. There is no necessity for a buffer stock of bodies. Part of the marshalling area for power units is shown in Fig. 1.

Fig. 6. One bay of the marshalling area for small parts

The marshalling conveyors are King "Dual-Duty" twin rail conveyors fitted with indexing units so that when the leading trolley with its load is released to the tunnel feeding conveyor, all the other units move up in sequence.

Releases from the marshalling conveyors are controlled by a Hollerith punch card system. In effect, the Hollerith machine acts as a master switch for correlating deliveries from the marshalling conveyors with the progress of chassis along the assembly conveyors.

From the marshalling conveyors, the various units are automatically transferred to the tunnel conveyors, see Figs. 2 and 3. These are King "Dual-Duty" conveyors which run at high speed and have widely spaced pusher dogs so that only a small number of units is in the tunnel at any one time. The return runs of these conveyors also pass down the tunnel. Incidentally, bodies are carried in specially designed quick-release cradles, the same type of cradle being used for all bodies. Power units are fitted with specially designed eyebolts and are suspended on hooks which disengage automatically after the

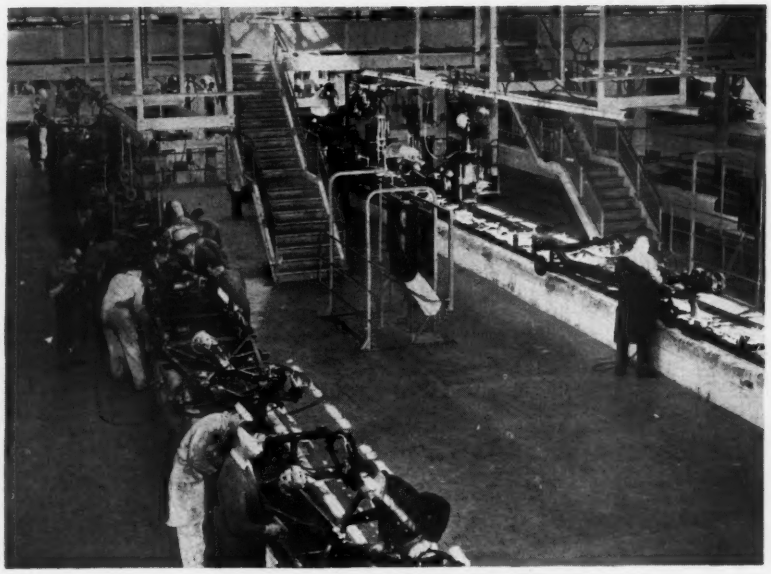


Fig. 7. General view of chassis build convevor lines

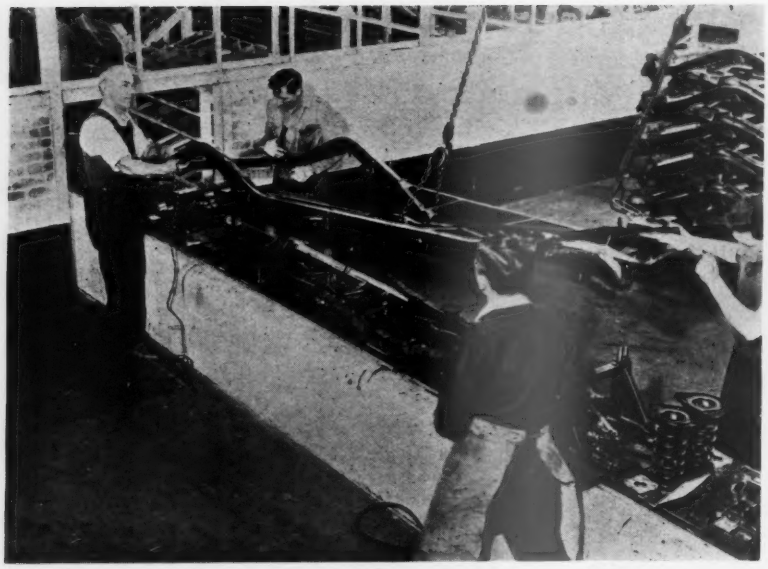


Fig. 8. Positioning a frame on its cradle on the slat conveyor

engine has been mounted in the chassis.

Leading off the tunnel are three pits from which the assembly tracks are supplied. That is, there is one pit for rear axle and front suspension units, a second for power units and a third for bodies. In each pit there are three elevating conveyors, one for each assembly track. From the tunnel conveyors the units are automatically transferred to the indexing conveyor feeding to the appropriate track, see Fig. 4. Each of these indexing conveyors will hold three units. As soon as there are three units on, the tunnel conveyors are automatically brought to rest. As soon as the leading unit is transferred to the elevating conveyor, the others move up one position and the tunnel conveyors start again to make further deliveries.

From the feeder conveyor the unit is automatically transferred to the travelling load bar of a vertical elevator for transfer to the assembly track, Fig. 5. The operation of these elevators raised some difficult problems. Perfect synchronization with the assembly track must be maintained. The complete cycle of delivering the unit to the mounting station and then returning the travelling load bar to the down position ready for the next unit, must be completed in about two minutes. It is essential that there shall be almost no swing during the travel to the mounting position. This problem was finally solved by fitting Heenan and Froude dynamatic electronically-controlled slip couplings. These very sensitive couplings allow the optimum rates of acceleration and de-acceleration to be used.

The operating cycle for the elevator is initiated by an electrical signal through a limit switch contacted by the assembly track when the chassis is approaching the station at which the unit is mounted. Suitably placed limit switches contacted by trips on the elevator chains control the speed of travel. The Heenan and Froude coupling gives virtually instantaneous response to these signals and such exact timing that the power unit is lowered into exactly the correct position on the chassis.

The assembly factory

Essentially, the assembly factory comprises a stores area, a marshalling area and an assembly area. Between them the stores and the marshalling areas hold two days' supplies of small parts, all of which are received in pallets. These parts are issued in unit sets from the marshalling area, see Fig. 6. All the conveyor tracks for chassis building start in the marshalling stores and the loading of unit sets on to the slat conveyors is carried out to a planned sequence so that at each station the necessary parts are in the position most convenient for the operator. This has two advantages. First, the operators never need to move to pick up the parts, and second, since the loading sequence is constant the assembly operators soon develop a rhythm that helps them to maintain pace with the conveyor without fatigue.

As the conveyor emerges from the marshalling stores through a partition wall into the assembly area shown in Fig. 7, the chassis frame already lifted from the stock pile on King electric pulley blocks, is lowered into position for the assembly operations to start, see Fig. 8. At the early stations, shackles, rear springs and other details are assembled to the chassis before it reaches the stage at which the rear axle and front suspension units are elevated from the pit to the track.

The rear axle is dropped into a cradle between the chassis frame side members. It lies at right angles to its normal position, but when the chassis reaches a certain point in its travel, the rear axle is automatically turned through 90 deg. to be brought into the transverse position beneath the road springs. Specially designed pneumatic equipment is used in the assembly to the chassis of rear axle and front suspension units. That for the front suspension is shown in Fig. 9. Certain fixing bolts cannot be inserted until two strong coil springs have been compressed. Compression is effected by means of two air-operated clamping units, one on each side of the track. These units are carried on wire cables attached to counterbalance suspension units so that they can easily be pulled down to the working position and return automatically to the off position when released. The suspension units are

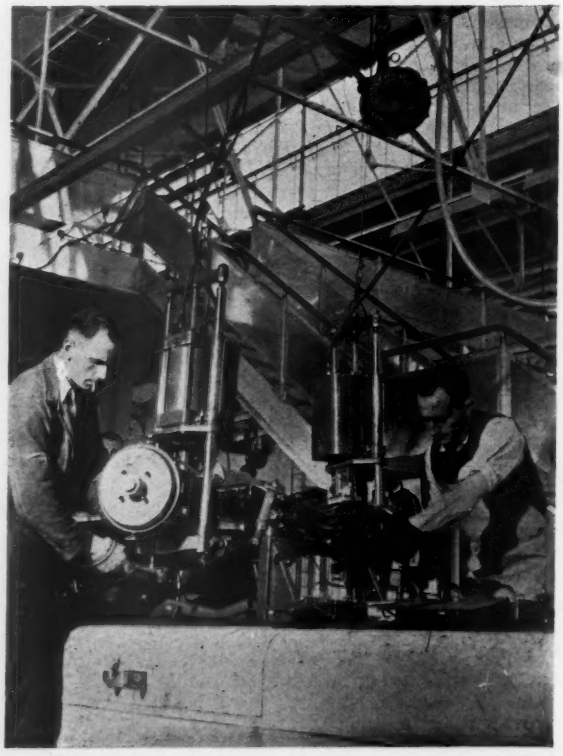


Fig. 9. Mounting the front suspension unit. Special pneumatic tools are used to compress the coil springs

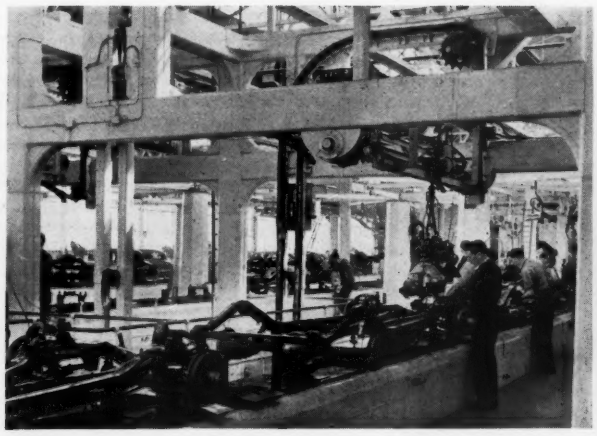


Fig. 10. Engine mounting station. The power unit is automatically lowered into position

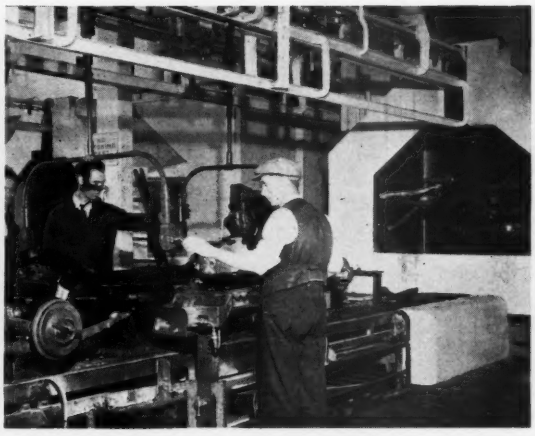


Fig. 11. End of the chassis build conveyor and the entrance to the electrostatic spray booth

carried on an overhead runway so that they can travel along with the chassis. Adjustment of the front suspension to give correct tracking of the wheels is carried out on the conveyor by means of a device clamped to the brake drums and incorporating optical means for checking the setting.

Great use is made of pneumatic tools for tightening bolts, nuts and screws. Desoutter heavy-duty, high-pressure tools are used throughout. Some are standard single tools, but others are specially developed multi-head tools arranged for tightening as many as four bolts at once. Special attention has been paid to positioning the tools to give optimum operating convenience. When not in use some tools are placed in special rests at the side of the conveyor, others are carried by suspension units similar to those shown in Fig. 9, while yet others are carried on arms projecting horizontally from a central hub that can be rotated to bring the required tool into the working position.

At succeeding stations other details such as the propeller shaft, the brake mechanism and the cable harness are assembled to the chassis before it reaches the station at which the power unit is mounted in position. Shortly before this station is reached, a limit switch is tripped to start the elevating conveyor that brings the power unit from the pit ready to lower it on to the chassis, see Fig. 10. So accurate is the synchronization that the engine is lowered exactly into position without any work on the part of the operators. Engine mounting and the completion of other details concerned with the clutch control and the steering complete the chassis build on the slat conveyor.

Fig. 12. Oiling station where all chassis lubrication is carried out under air pressure

At the end of the slat conveyor there is an air-operated mechanism which automatically raises the chassis clear of the track for transfer to a Dual-Duty conveyor running through an electrostatic spray booth and a drying oven, see Fig. 11. Any necessary masking is carried out before the chassis leaves the slat conveyor. In the spray booth the chassis passes through an electrode system charged to a potential of 150,000 volts. In its passage it meets a finely-atomized paint mist which becomes highly charged with electricity and is thus drawn on to all the surfaces of the work to an evenly distributed coating. This electrostating spraying eliminates the need for operators, gives a better result and effects very great savings in the amount of paint used.

On leaving the drying oven, the chassis is transferred from the Dual-Duty conveyor to a slat conveyor for passage through a lubricating section for which the equipment has been designed and constructed by C. C. Wakefield and Co., Ltd., see Fig. 12. In this section the engine sump, the gearbox and the rear axle are filled with oil, the hydraulic brakes are filled and bled, and the grease nipples are charged. The oil store is sited conveniently in relation to the lubrication section. In it there are six main tanks, each of 1,300 gallons capacity. They contain engine oil, gearbox oil and extreme pressure lubricants. There are also smaller tanks of 150 gallons capacity for special lubricants and for hydraulic brake fluids.

To give them the fluidity necessary for easy handling and to enable them to penetrate new, closely-fitting bearings, the engine, gearbox and axle oils must be brought to 150 deg. F. Each of the main tanks is heated by thermostatically controlled steam coils and the oil mains are lagged to prevent heat losses. From the main tank the oil is pumped into an accumulator, each tank being provided with two electric pumps. these are completely automatic in operation, and cut in and out to meet the factory requirements without any

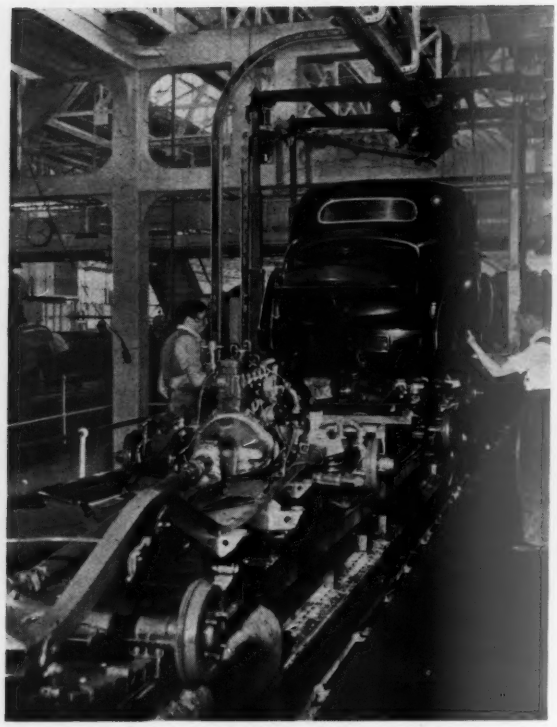


Fig. 13. Body being lowered into position from the elevating conveyor

human supervision. In the accumulator the oil is subject to the pressure of compressed air. It is this pressure that forces the oil to the nozzle of the hose at the lubricating stations. The pressure is automatically regulated. It differs for different oils. For example, for very low viscosity fluids such as are used in hydraulic braking systems, 25 lb. per square inch suffices, but for axle oils 220 lb. per square inch is necessary.

A main Wakefield lubricating station spans each track. The lubricating hoses are carried on a carriage that travels on a runway above the assembly conveyor. This carriage is about 8 ft. long and the overall length of the main stations is 33 ft. 4 in. An ingenious method is used for synchronizing the travel of the carriage with that of the track. There is a wishbone shaped towing arm on the carriage, which descends at the correct instant to engage a driving member on the track and immediately in front of the chassis. This necessarily ensures that the carriage moves at exactly the same speed as the chassis.

At the main lubricating station, two operators, one on each side of the track, apply guns to each grease nipple to give a charge of Castrol Hi-press. During the passage through the lubricating station, 7½ pints of oil are automatically metered to the engine sump, 2¾ pints of gear oil to the gearbox and 2½ pints of extreme pressure oil to the rear axle. When the filling is completed the filler caps are screwed home and given a dab of red paint to indicate that the unit has been oiled.

The apparata for metering the various oils are housed in cabinets alongside the track. Essentially, each supply is controlled by a double-acting cylinder, jacketed to maintain the oil temperature. Within the cylinder is a piston with an adjustable stroke. The stroke is set by a handwheel and the capacity can be read directly from an engraved slide bar. Once the correct setting has been made, the delivery of the correctly metered amount is completely automatic.

When oil under pressure from the accumulator enters the upper end of the cylinder, it forces the piston down and drives out the oil in the lower part of the cylinder. The valve gear then switches the oil supply to the lower end of the cylinder to force the oil above the piston out to the lubrication station. The cycle is repeated for each chassis, and every stroke delivers sufficient oil for one chassis. There are, of course, separate metering cylinders for engine, gearbox and rear axle oils.

When the chassis has completed about three-quarters of its journey through the lubrication station, a trip on the side of the hose carriage engages an electric switch to cause an alarm bell to ring to warn the operators to lift the hoses clear. Once the nozzles are clear of the filler cups, the hoses are automatically reeled. A few inches further along, the trip engages an air switch which brings into action compressed air mechanism for returning the carriage at the end of its travel. At the end of the run another air control actuates a pair of cylinders to lift the towing arm clear of the conveyor so that the compressed air piston can drive the carriage and towing arm back to the starting point to meet the next chassis.

The hose from the brake fluid reservoir also hangs from the carriage. While the fluid is entering the reservoir, the ends of two plastic hoses are pushed over the bleeder nozzles on the brake anchor plates. Two operators slack off the bleed screws, give the brake pedal two jabs, remove the bleed hoses, and the brakes are ready for service.

Shortly after leaving the lubrication station, the chassis reaches the assembly station at which it meets the body, see Fig. 13. The body is delivered complete with petrol tank, steering column, bonnet and radiator. The spare wheel is in position and the four road wheels are carried in the luggage boot. Inside the body are all the parts and fittings necessary to complete the car. Immediately the body has been attached to the chassis the road wheels are fitted. This brings the assembly to the end of the second slat conveyor. The vehicle is then carried on its wheels on an elevated twin slat conveyor of such a height that men standing on the floor can work on the underside while men on platforms at conveyor height can also carry out other operations. This

Fig. 14. A completed vehicle leaving the assembly track

arrangement allows as many as nine men to work at one station.

In its passage along the mounting section, the car passes through a small Wakefield lubricating station which delivers a metered quantity of oil to the steering box. There is a station on each side of the conveyor so that both right and left hand drive cars can be serviced without the mechanics having to cross the track. A little further along, the petrol pipe is coupled up as the vehicle approaches a Wakefield equipped petrol station. This draws fuel from underground storage tanks outside the building. An operator opens the petrol tank filler cap and inserts the nozzle of the hose from the pump. This pump is set to deliver one-half gallon to each vehicle. When this quantity has been delivered, the pump is shut off automatically. As the car moves forward, a trip on the side of the track engages an air switch, and compressed air mechanism releases the pump lock, re-setting the pump to deliver half-a-gallon to the next car.

During the passage along the elevated twin slat conveyor the radiator is filled This is done by means of Wakefield equipment of the type used for filling the engine sump. As the cooling system takes 12½ pints and the maximum delivery of the double-acting cylinder is only 10 pints, the cylinders are duplicated, one delivering 6 and the other 6½ pints. This arrangement has been adopted to allow one size of cylinder to be standardized throughout the factory. At the end of the line, see Fig. 14, the twin slat conveyor falls to floor level so that the car may be started and driven away under its own power.

There are almost sixteen miles of conveyor tracks in connection with the assembly factory. These, together with

Fig. 15. The control room for all electrical supplies used in the assembly factory

electric hoists and electric tools, are joined by some 100 miles of cable involving 14,000 separate connections direct to the control room shown in Fig. 15. Should any fault occur in any part of this system, its location is immediately indicated by means of coloured buttons or flashing lights. This allows remedial action to be taken immediately.

If optimum output is to be obtained, it is necessary to provide the best possible working conditions, and there is no doubt that this matter has been studied as carefully as has the question of equipment. For example, some 60,000 square feet of glass, all arranged vertically, is used in the roof and walls to ensure maximum daylight. This, together with natural cross ventilation and efficient insulation, ensures warmth in winter and coolness in summer.

In conclusion, it is worth stressing again that this assembly factory forms an essential part of a comprehensive plan for integrating the whole of the productive processes in the Longbridge works. It is probably the most ambitious assembly project that has ever been carried out, and it certainly embodies techniques of automatic conveying of materials and of stock control that are much in advance of anything hitherto attempted.

FATIGUE FAILURES

IN "*Fatigue Failures Are Tensile Failures*," by J. O. Almen, published in *Product Engng.*, March, 1951, it is stated that compressively pre-stressed metal forms a barrier which prevents the propagation of fatigue cracks. The article quotes several examples in which intentionally or unintentionally imposed compressive residual stresses resulted in a considerable increase of fatigue strength.

When gear teeth are ground, a thin surface area may expand beyond the elastic limit. On cooling, tensile stresses may produce surface cracks. These are not dangerous, however, if the residual stress is converted from tension to compression by the cold working of contacting teeth. As the root portions of teeth are not subjected to cold work, a method has been developed for grinding the tooth flanks only. Shot peening increases fatigue strength whether or not the teeth are ground. Beneficial compressive stresses can be produced by carburizing or nitriding surfaces which tend to expand but are restrained by the underlying core. A similar effect is achieved by flame and induction hardening, except that compressive stress is partly counterbalanced by a surface tensile stress similar to the grinding stress. Local heat of welding establishes a compressive barrier between a thin surface layer and the core in tension.

Fatigue failures can result from the application of a nominal compressive stress, as in the case of a Belleville Spring. Central parts of the spring yield under a high external load and become residually stressed in tension after removal of the load when the spring elastically returns to its original conical form.

Rims of railway carriage wheels are cast in a chill or quenched for the purpose of hardening the rolling surface. This operation develops residual tensile stresses in the plate and hub with corresponding residual compressive stress in the rim, and thus prevents the propagation of rim surface cracks caused by heat generated on braking. Residual compressive stress in the rim may be lost through severe and prolonged braking if, owing to additional stress, plastic yielding occurs. Subsequent cooling will then reduce the original compression, or even set the rim in tension.

Pre-stressing is not beneficial when applied load greatly exceeds the elastic limit involving an immediate loss of the protective compressive stress, and the presence of bruises and cuts caused by peening reduces fatigue strength below that of non-prestressed material. (*M.I.R.A. Abstract* 5346).

New Directorship

Following on the recent death of G. Geoffrey Smith, Arthur B. Bourne has been appointed a Director of Iliffe & Sons Ltd., proprietors of *The Automobile Engineer*. Mr. Bourne has been with the firm since 1926, and has been Editor of *The Motor Cycle* for the past twenty-three years.

CAM FEED DRILLING

Additional Equipment for Herbert All-electric Machines

THE method of unit construction adopted by Alfred Herbert Ltd., Coventry, in the manufacture of all-electric drilling machines permits the erection of multi-spindle machines with varying combinations of spindles to suit a wide range of requirements. This range has recently been increased by the introduction of cam feed heads which can be so arranged that the normal functions of machines fitted with this equipment are performed automatically. The machine operator need only load and unload the component.

On high production work the performance of Type C drilling machines can be greatly improved by the addition of cam-feed heads. When used with suitable jigs or fixtures they allow idle times to be cut to an almost irreducible minimum. Figs. 1 and 2 show one of these machines so equipped for drilling fourteen equally-spaced holes in the scraper ring groove of a piston. This type C manufacturing machine can be supplied with any single speed from the range of eleven speeds between 72 and 2415 r.p.m. that are available.

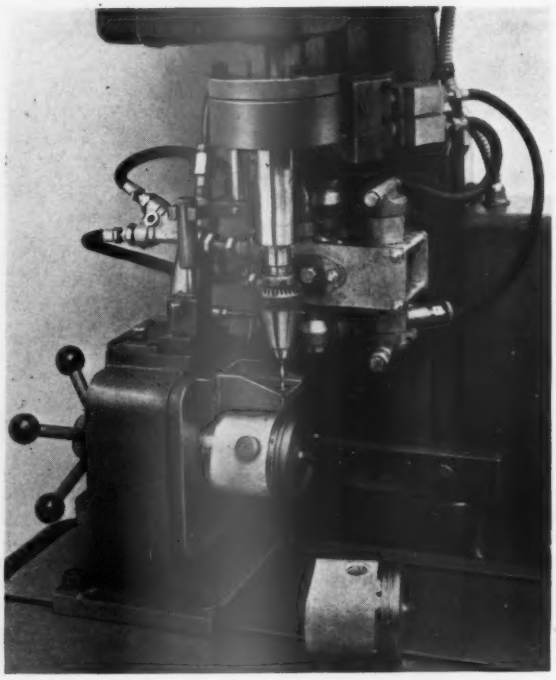


Fig. 1. A special fixture on a Herbert cam feed drilling machine.

On the machine illustrated the feed shaft is driven through reduction gearing by the main spindle. In this case the rates between the two are 2·745 : 1. The motions of the spindle are controlled by two cams, one of which can be seen in Fig. 2. This cam controls the downward motion of the spindle. It can be designed to incorporate such variations in the speed of motion as are required for rapid approach to the work, normal feed and dwell. In drilling deep holes on hand-feed machines it is common practice to clear the drill of swarf in order to avoid drill breakage. This motion can also be reproduced on the cam so that it occurs automatically. A smaller cam, bolted to the inner side of the large cam, controls the spindle withdrawal. The shaft on which the cams are mounted is driven by the feed shaft through pick-off gears and a worm and worm wheel mechanism. The pick-off gears provide a means of varying the camshaft speed and therefore the rate of feed.

A timing device is fitted to the right-hand end of the camshaft to stop the machine automatically at the completion of each cycle. It consists simply of an electrical limit switch which is tripped at each revolution of the cam by a dog mounted on the camshaft. Methods are available whereby the machine can be arranged to stop after completing a desired number of cycles. For these, electrical connections with the fixture are necessary.

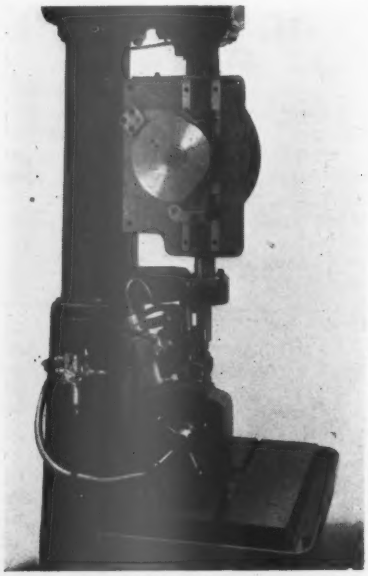


Fig. 2. The cam for controlling the downward stroke of the drill spindle.

The fixture used for drilling the piston is an air-operated indexing type. It is indirectly actuated by the motions of the machine spindle. Indexing is effected by Geneva motion through 14 positions and takes place in the following manner:—

At the top of the stroke, the spindle operates a trip valve mounted on the machine column thereby directing air into the outer end of an air cylinder and causing an inward movement of its piston. The air cylinder is secured to the top of the fixture into which a rack extension of the piston projects to mesh with a small pinion. From this pinion, movement is imparted to the driving plates and pegs of the Geneva motion by a pawl and ratchet mechanism. This form of transmission is necessary because, owing to the reciprocating motion of the spindle, the pinion rotates alternately in clockwise and anti-clockwise directions, whereas the rotation of the driving plate must be in only one direction. Thus on the inward stroke of the piston the pawl on the pinion engages the ratchet on the driving plate and pushes it through approximately 180 deg., thereby rotathe Geneva index plate through one-fourteenth of a revolution. At this point the driving plate is retained by a plunger under light spring pressure while the drill commences the downward stroke. A bar attached to the drill spindle also travels down and positively locates the Geneva index plate just before the drill starts cutting.

At the bottom of the stroke, another trip valve is operated to direct air into the inner end of the air cylinder to push the piston and rack outwards in preparation for the next indexing motion. This also rotates the pinion, but the driving plate is not affected, since the pawl merely rides over the ratchet teeth. Meanwhile the drill spindle again reaches the top of its stroke and the cycle of operations is repeated. To stop the machine automatically after fourteen cycles, a switch similar to that on the cam feed is fitted to the fixture. It is tripped at the end of each revolution made by the Geneva index plate. The two switches are so wired that the cam-head switch does not operate until that on the fixture has also been tripped.

THE "RIVALOY" ALL-METAL BODY

Pre-fabricated Coach Bodies Exported as Knocked-down Units

WITH the increasing development of export business in recent years, the need has arisen for the supply of bodies in the knocked-down condition. One of the largest suppliers of coach bodies in this market is Saunders-Roe (Anglesey) Ltd., who recently demonstrated the assembly of their "Rivaloy" all-metal body on a Leyland "Royal Tiger" chassis. A high rate of output and exact interchangeability of components and detail parts is ensured by the very complete jigging and tooling provided for production. The bodies are built from the following main units :— sides, floor, roof, front end and rear end, these major assemblies being joined together in general by means of back-to-back aluminium alloy channel members bolted together with floating anchor nuts located on one channel.

The body sides are formed from H-section aluminium alloy pillars and steel Z-section waist, seat and floor members. Panel drumming is prevented by the use of felt-lined steel members, and T-section aluminium alloy stub pillars are provided at certain locations. To this basic structure are riveted the stress bearing steel interior panels, and steel crib rails are riveted to the floor rail members. Plugged Chobert rivets are used to attach the exterior panels which are lapped at the pillar stations and floor rails. The aluminium alloy channel section cant rails are bolted back-to-back with the roof cant rails, thus forming built-up H-section members.

Double skin roof

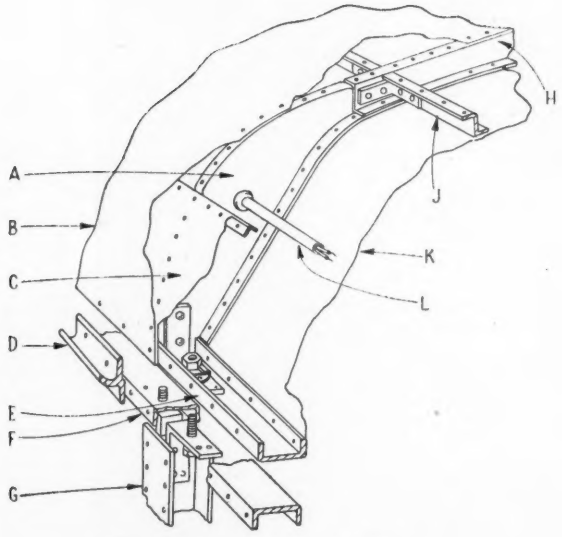
Aluminium alloy panels form a double-skinned roof, the exterior panels being solid riveted and the interior panels Chobert riveted to the structure. This is formed of steel Z-section roof sticks, longitudinals and corner gussets, with side channels of aluminium alloy and full length steel cant plates riveted to the side channels and corner gussets. A layer of insulating material is placed between the two roof skins.

Structurally similar to the sides with aluminium alloy pillars and steel rails crib rails and panelling, the rear end has a double skinned dome of steel sheet with insulation similar to that of the roof. The front end, however, is built up from H-section corner pillars, Z-section rails and angle crib rails and steel diaphragms. The internal steel panelling is solid riveted to this structure, but the external panelling is Chobert riveted in six sections. Thus easy repair is ensured in the case of minor accidents. Sheet steel is also used for the windscreen pan and front canopy, which are attached by solid rivets.

Bolted to the chassis members are timber packings to which are screwed timber boarding longitudinally laid to form the floor, with additional transverse timber reinforcing at the seat locations. Double zinc-coated steel is used for the wheel arches while the step wells are made from aluminium alloy having chequer treads.

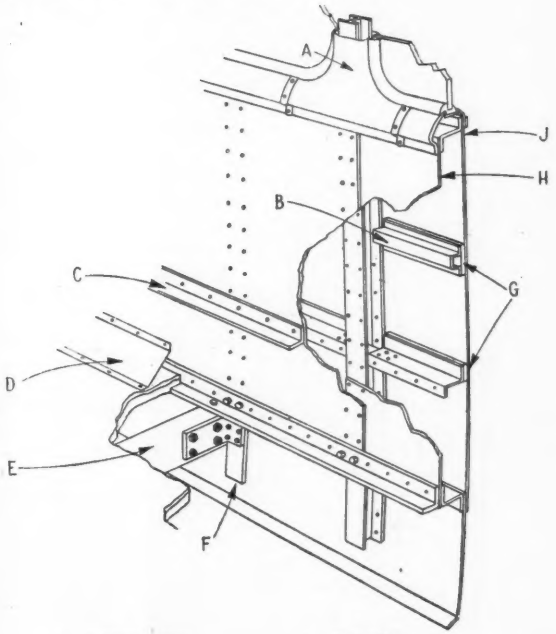
Body mounting

The body is mounted on the chassis by means of steel connectors bolted to the steel outriggers and picking up with the main side frame pillars or with a stub pillar in cases where the outriggers do not coincide with a main pillar. At the front the end section is attached to the chassis by brackets connecting the corner pillars to the front outriggers and at two points along the crib. The rear end is attached by a bolted connection between the rear crib rail and an angle section bolted along the chassis rear end member.



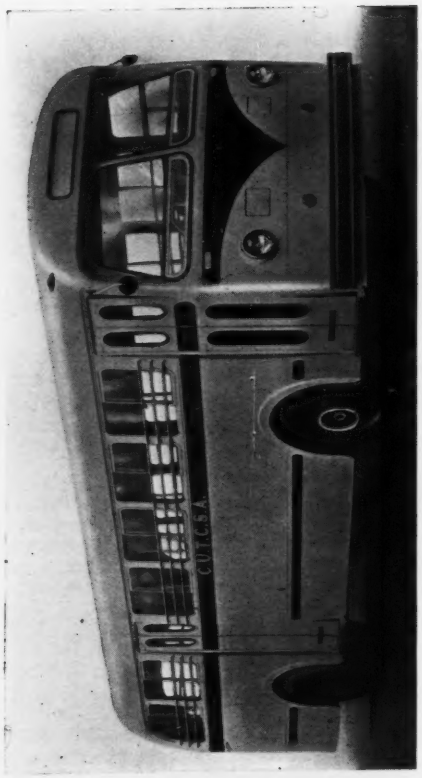
A. Roof diaphragm
B. Outer panel
C. Full length cant plate
D. Gutter mould
E. Cant rail
F. Side frame cant rail
G. Pillar
H. Roof stick
J. Longitudinal member
K. Lining panel
L. Wiring conduit

Joint between roof and side frame



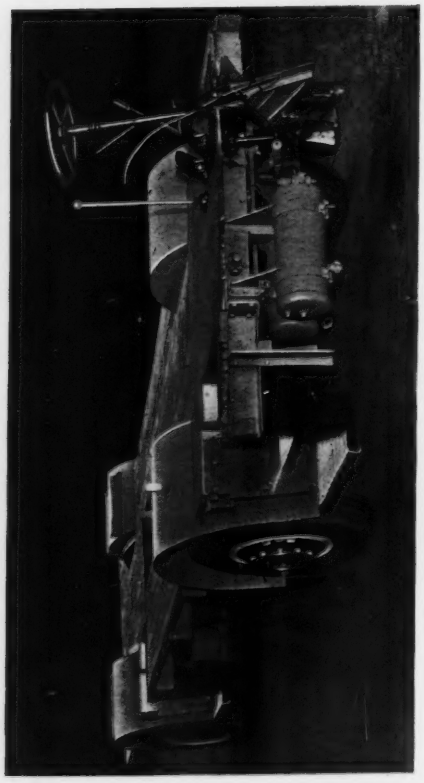
A. Finishing shroud
B. Anti-drumming rail
C. Seat rail
D. Covering panel
E. Chassis outrigger
F. Stub pillar
G. Felt padding behind rails
H. Inside stress panel
J. Outer skin

Side and floor construction



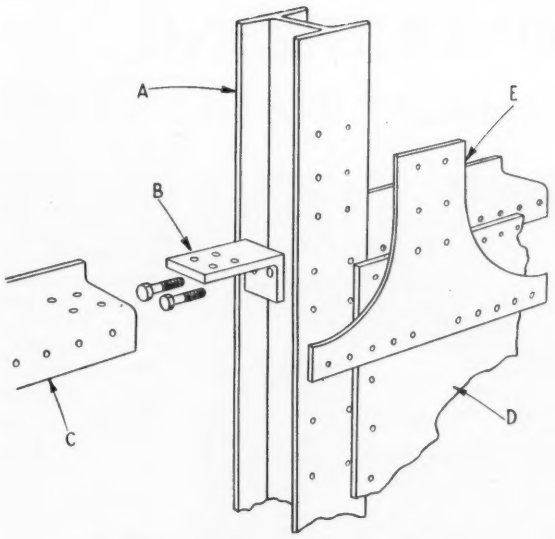
Forty-seater "Rivaloy" body

Interior of forty-seater body



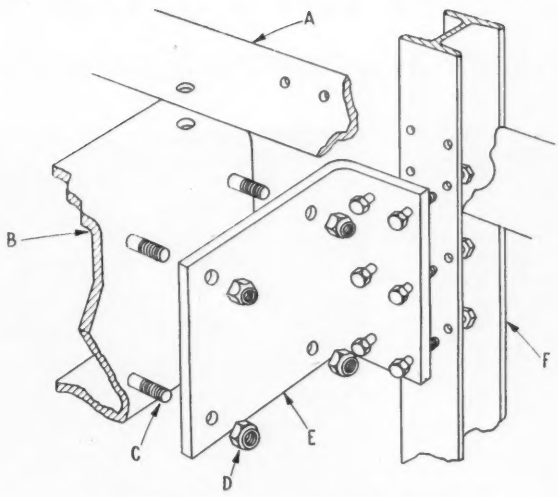
Chassis ready to receive bodywork

Assembly of side frame on chassis



A. Pillar
B. Extruded angle cleat
C. Waist rail
D. Inside stress panel
E. Gusset

Pillar and waist rail joint



A. Crib angle
B. Chassis outrigger
C. H.T.S. bolts
D. Aerotight nuts
E. Angle connection
F. Pillar

Details of pillar to chassis outrigger joint

General construction

The design of this body is based on the principle that any one part should be used for as many purposes and in as many positions as possible, in order that first cost should be low, overseas assembly from completely knocked-down parts should be simple, and the stock of spare parts to be held by the operator should be kept to the minimum. The sub-assembly of all parts is arranged so that the main body units are self-jigging. Thus unit assemblies, that is to say the floor, near- and off-side frames, front and rear ends and roof, can be assembled with unskilled labour quickly and with very simple jigging. As delivered in the knocked-down condition, all units are completed except for certain finishers before being assembled as a body, and can be finish painted if required.

The roof, for example, is skinned inside and outside and is complete with insulation, electrical wiring, lamps, ventilators and grilles, mouldings, bell cords, etc. It is claimed that this permits the knocked-down body to be assembled overseas in less than 160 man hours, provided that the labour employed is reasonably familiar with the general type of construction. Further, since the units are complete before assembly, the number of bodies in course of erection at any one time is considerably reduced, thus saving floor space. Finally, once the units are assembled together, only items such as seats, wheel arch valances and certain finishers need to be installed before the vehicle is ready for service.

Maintenance and repair has been carefully studied in the design, and the methods developed permit minor damage to exterior panelling to be corrected without disturbance of the interior of the body. Further, damaged pillars can be easily repaired, while waist, floor and other side frame longitudinals which are of steel can be quickly welded or replaced. Complete units such as front or rear ends can also be quickly removed and replaced owing to the full interchangeability of all components. The vehicle will therefore be out of service for the minimum possible time. Repair tool kits for carrying out panel replacement are provided with each contract.

EXAMINATIONS IN MOTOR BODY WORK

The City and Guilds of London Institute has recently announced the adoption of revised syllabuses for its Final grade examinations in Motor Body Work. As the revision of the Intermediate syllabus was completed in 1950, this means that the entire scheme has now been brought into line with modern developments in the industry.

Whereas the old scheme provided for Intermediate and Final examinations in both Private Motor Body Work and in Commercial Motor Body Work, but did not cater for large-scale production, the new scheme has a common Intermediate examination with three separate Final examinations, as follows:

Motor Body Work for Small-scale Production.
Motor Body Work for Commercial and P.S. Vehicles.
Motor Body Work for Large-scale Production.

The curriculum for each course has three main sections—Technology of Motor Body Work (including drawing), Science and Calculations, and a Scheme of Practical Work. The introduction of the syllabuses on "Science and Calculations" and of the "Scheme of Practical Work" brings the examinations in this subject into line with current practice in other City and Guilds examinations in mechanical engineering subjects. This will probably present technical colleges with some difficulties in organizing the necessary classes, particularly in connection with the practical work in relation to large-scale production, in which co-operation with industry will be essential.

A pamphlet containing the revised regulations and syllabuses is being prepared and may be obtained (price 6d.) from the Department of Technology, City and Guilds of London Institute, 31, Brechin Place, London, S.W.7. Copies of the question papers for past examination—including the Intermediate examination (revised scheme) of 1951—may be obtained from the same address, price 6d. per paper.

SPRAY GUN BONDERIZING

A Pyrene Cold Pre-treatment Process

AN interesting addition has recently been made to the range of metal pre-treament and rustproofing processes marketed by The Pyrene Company Ltd., Great West Road, Brentford, Middlesex. It is a Bonderizing development that uses a cold-processing solution which is applied to sheet metal by means of an ordinary paint-type spray gun. This development makes available the advantages of Bonderite pre-treatment to specialist and small scale manufacturers whose production circumstances do not warrant the installation of orthodox Bonderizing plant. The new process is therefore, not an alternative to the established systems in current use, but is an extension of them.

Spray gun Bonderizing at the works of Singer Motors Ltd.

Spray gun Bonderizing, which is intended for use only on good quality sheet steel, does not call for the provision of special equipment. Nor does it necessarily demand the provision of space reserved solely for the work. Where, however, there is a steady flow of production to be treated, it is advantageous to have a spray booth allocated to the spray gun Bonderizing operation. Preferably, the spray booth should be of the water-conditioned type.

The phosphate coatings produced on steel surfaces by this process possess the same paint-bonding and corrosion-resisting qualities as those obtained from orthodox Bonderite treatments, and a similar operation sequence is employed. Preparation of the steel surface, which must be thoroughly de-greased, is the first stage in spray gun Bonderizing. For this purpose the use of turpentine substitute (white spirit), trichlorethylene or other suitable solvent is essential. Normally, wiping with turpentine substitute followed by dry wiping is employed. If any rust exists on the surface, it must be removed. For removing rust, the Pyrene product "Preperite" is recommended for "spotting in" on rusty areas.

In treating the work, spray gun Bonderite solution, at the concentration in which it is supplied, is sprayed on to the cleaned surface. An air pressure of up to 40 lb. per square inch is recommended. The air pressure is not, however, critical. Processing is carried out at room temperature. If use is made of a pressure feed pot, the pot must be lined with an inert material such as vitreous enamel, glass or stainless steel. Application of the solution should be made as rapidly as possible consistent with thorough and complete wetting of the whole surface to be processed.

After application the solution is allowed to remain in contact with the steel surface for a period of 2½–5 minutes, after which the surface is thoroughly rinsed with cold water, normally applied by hosing. Hot water should not be used for rinsing. The work must then be thoroughly dried off. For this purpose, oven drying is to be preferred but compressed air can be used quite satisfactorily. After being dried off, the surface obtained is chemically cleaned and in the correct condition for the application of the final finishing sequence.

As all Bonderite coatings are highly absorptive, paint should be applied as soon as possible after the drying off operation. Painting can be carried out with full confidence that the spray gun Bonderite coating will impart the same degree of increased adhesion and improved durability to the final finish as are obtained from the orthodox Bonderizing pre-treatment processes. The process in use at the Birmingham Works of Singer Motors Ltd., is shown in the accompanying photograph.

Plastic Protective Coating

THE development of a new air-drying plastic protective coating has been announced in the May 1951 issue of *The Engineers' Digest*, which can be used to protect tanks, tank cars, pipes, fittings, structural steel parts and chemical processing equipment against attack by splash, drip and spray from corrosive chemicals, corrosive atmospheres, weathering and rust.

This plastic combines high film flexibility with good adhesion and impact resistance. A claim is made that it will not chip or crack and that it can be used on steel, aluminium, concrete, hardwood or composition board. It is applied by spray, allowing one hour drying time between coats and twenty-four hours drying time after the final coat. It requires no primer and can be produced in a variety of colours. Since it contains no chemical plasticizer which would be subject to leaching out or hardening, the coat retains its flexibility throughout its service life. A sharp instrument will cut through the film, but the break can be repaired easily by touching up with brush or spray. Metal should be grit- or sand-blasted before the coating is applied; cement should be etched with hydrochloric acid, while wood and composition board need rough sanding.

ENGINE INDICATING

The C.A.V.-Ricardo Photoelectric Pressure and Movement Gauges

IN a description of the modified Hartridge test stand used in the Ricardo laboratories, published in the *Automobile Engineer* for June 1949, this gauge was briefly referred to and its use and method of operation explained. It has now been possible to obtain details and illustrations of the gauge itself, together with the units of a typical installation. The gauge is designed for the measurement of rapidly changing pressures such as those occurring in the fuel injection system and also of the movement of the needle. These values are converted into proportionate electric signals, that can be amplified as required. With the aid of suitable electronic equipment a graphical picture of the varying quantity can be produced on the screen of a cathode ray tube against a suitable time base.

The majority of experimental and testing laboratories have amongst their test apparatus a general purpose oscilloscope, which, in conjunction with the photoelectric gauge can be made to function as an engine indicator. Useful indicator diagrams have been obtained using these gauges with a number of well known makes of oscilloscopes. The gauges were developed in the C.A.V. Research and Development Laboratories in collaboration with Ricardo and Co. Engineers (1927) Ltd. Their general form will be seen from the illustration.

For registering pressure, a small diaphragm of 0·8 in. diameter is used, the thickness being 0·01 in. or more, depending on the pressure to be measured. To the centre of the diaphragm is soldered an inverted U-shaped metal piece, the centre part of which acts as the shutter of a small optical slit. The opening of this slit is increased by deflection of the diaphragm. Needle lift diagrams are

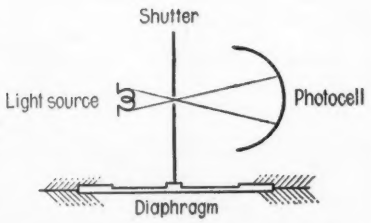


Fig. 1. Diagram showing principle of operation

obtained with a modified injector holder in which the needle motion is transmitted directly to the slit shutter.

The principle of operation

As shown diagrammatically in Fig. 1, the slit shutter controls the amount of light from the filament lamp which falls on the sensitive surface of the photocell, and therefore the current in the photocell. Increase of shutter opening increases both the intensity of illumination and the area of the photocell illuminated, but the total light flux is proportional to shutter opening so that there is a linear relation between shutter displacement and current in the photocell.

Electrical supplies to the lamp and photocell are as follows :—

	Lamp	Photocell
Voltage	6 V D.C.	90 V D.C.
Current	0·2 amp	3 μA
Stability* ..	±0·25%	±0·5%
Max. Perm. A.C. Ripple*	0·2%	0·25%

* These values are only intended as a guide. Actual values would be chosen to suit the application.

Output

The output is limited by the current which may be taken from the photocell, which is 2·5 μA. Fig. 2 is a typical curve showing the relation between shutter opening and photocell current ; it is linear for shutter openings greater than 0·004 in. corresponding to a current of 0·6 μA. The maximum output is thus a current change from 0·6 to 2·5 μA, that is, 1·9 μA for a shutter movement of 0·005 in. In the pressure gauges the shutter is set to an opening of 0·004 in. with the diaphragm undeflected so that the output is linear over the full working range of the diaphragm. Fig. 3 shows a typical curve for relation between pressure and photocell current.

In the needle lift gauge the slit shutter has been replaced by a plunger (the moving member) in a coaxial cylinder in order to provide the calibrating facility. In this case displacement *closes* the aperture. The relation between shutter opening and photocell current is linear for openings from 0·5 mm to 1·35 mm (see Fig. 4). By adjusting the lamp brightness the full photocell output, 2·5 μA, can be obtained at a lamp voltage of 5·0 V for plunger travels down to 0·1 mm. Whenever possible, however, the gauges should be operated with the current when the shutter is in the rest position (i.e., open) at less than 2·5 μA (half this value is desirable), since the photocells will then have longer lives.

When a gauge is to be connected to a voltage amplifier preceding a cathode ray tube it is desirable to take the output from a resistor in the photocell circuit not exceeding 100,000 ohms so that the full scale output is a voltage change of 0·19 V.

Frequency response

The frequency response of the photoelectric head may be limited by the characteristics of the photocell or by the time constant RC of the output circuit consisting of the output resistor and the stray capacitance across it. With the length of low capacitance cable provided with the

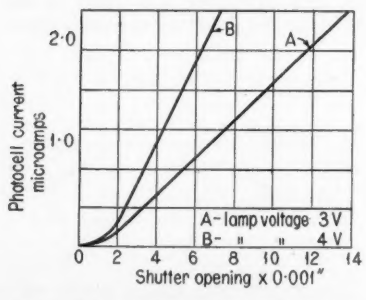


Fig. 2. Relation between shutter opening and photocell current

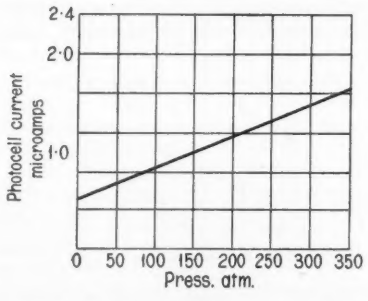


Fig. 3. Relation between pressure and photocell current

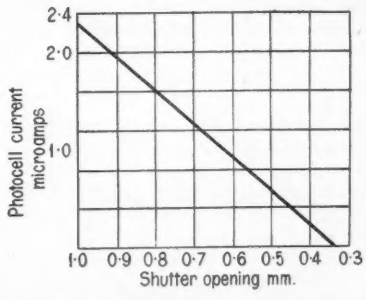


Fig. 4. Relation between shutter opening and photocell current on needle lift gauge

Fig. 5. Typical pressure-type gauge. Overall height 150 mm.

gauges and a 100,000 ohm output resistor the time constant is of order 10μs. A sudden displacement of the shutter will then be reproduced as output voltage change in such a way that 90 per cent of the final value is attained in 100μs, i.e., in about 2 deg. of rotation of an engine at 3,000 r.p.m.

Improvement in frequency response cannot be obtained by reduction of the output circuit time constant since the gas-filled photocell used does not respond in shorter times. The cell, however, may be replaced by the corresponding vacuum cell, but it has only about one-tenth the sensitivity. An increase of output voltage may be obtained at the expense of poorer frequency response by using a higher output resistance. The time taken to attain 90 per cent of the final value for a sudden change of input is proportional to the time constant, RC, of the output circuit.

The overall frequency response of a gauge may be limited by other factors. In pressure gauges the diaphragm will not respond in a time much less than its natural period and if it is underdamped the time may be several times the natural period. The transmission properties of the passage leading to the diaphragm cavity may cause delay and distortion. In movement gauges, a long metal rod connecting the shutter to the part it is wished to observe may fail to transmit the motion faithfully.

Diaphragms are available for both pipeline pressure and cylinder pressure gauges as follows :—

Pressure lb/in²	Thickness in.	Natural Frequency c/s
100 ..	.. 0·010 ..	.. 5,450
1,000 ..	.. 0·035 ..	.. 19,400
3,000 ..	.. 0·062 ..	.. 32,600
6,000 ..	.. 0·082 ..	.. 40,000

Application

Gauges are available for measuring fuel line pressure, Fig. 5, injector needle lift, Fig. 6, and cylinder pressure. Photographic records of typical cathode ray tube traces from these instruments are shown in Figs. 7, 8 and 9, which are self-explanatory. The photoelectric head common to both the pressure gauges is available separately for other applications requiring the measurement of small displacements.

Both types of pressure gauge may be obtained with special needle valves for rapid static calibration. They are arranged so that the diaphragm can be connected to the pressure to be measured, the atmosphere, or to an additional pressure connection to which a known static calibrating pressure can be applied.

It is intended that a pressure phenomenon having been recorded, zero pressure and a fixed calibrating pressure shall be recorded at once on the same film so that drift errors shall be minimized. The pressure to be measured is sealed off from the rest of the apparatus during calibrating which can therefore be done without interrupting the running of the engine or pump being tested. Diaphragms may be changed in order to alter the range of a gauge. They are supplied with the moving part of the shutter attached, and no special skill is required to change a diaphragm and reset the shutter.

It may be found that with the valve connecting diaphragm chamber to test pressure closed, the oscillograph may still show a small deflection. This

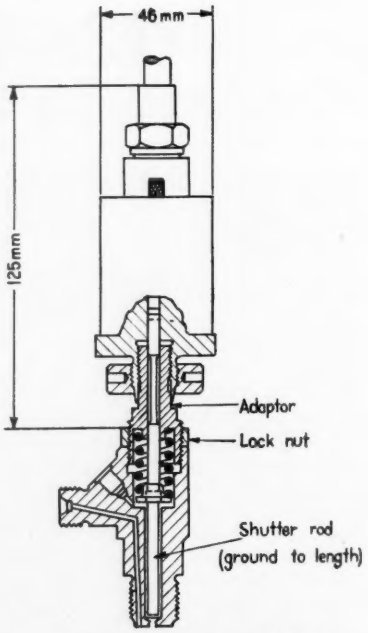


Fig. 6. Injector needle lift gauge

condition represents a complete blank test and the deflection obtained is a measure of the extent to which the normally recorded figure is spurious on account of mechanical vibrations to the lamp or shutter. In the presence of violent mechanical vibration, as in pipelines, the spurious deflections may be as great as 2 per cent of the normal figure, but should not exceed that value.

The photocells used are stated to withstand operation at 80 deg. C and no trouble due to overheating has yet been experienced. If, however, in any particular case this temperature is likely to be exceeded, cooling should be provided, by water jackets round the connecting passage for instance. Further possible causes of error are cyclic fluctuations of the temperature of the diaphragm or moving part of the shutter, and variation with temperature of these components between recording the observed phenomenon and calibrating.

The needle lift gauge is calibrated by screwing down the whole of the photoelectric head on the micrometer

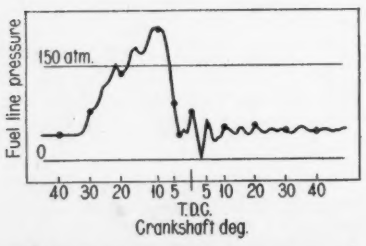


Fig. 7. Typical trace of fuel line pressure

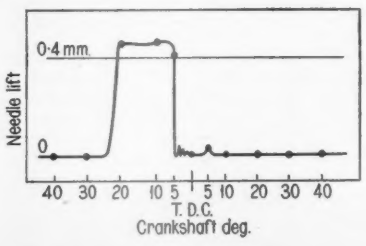


Fig. 8. Needle lift diagram

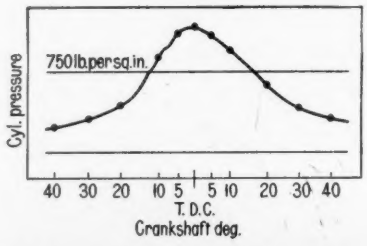


Fig. 9. Typical trace of cylinder pressure

screw. Uninterrupted calibration lines can be obtained if the engine under test is stopped; or alternatively, since the displacement is zero for most of the record, the calibration may be made without stopping the engine and results in a displacement of the whole record by a known amount.

When it is required to operate the gauge in conjunction with one of the well-known makes of oscillograph, it will be found to be of most value when used with an oscillograph embodying direct-coupled amplifiers, since static calibration can then be obtained. To connect the gauge to the oscillograph an input circuit and supplies such as in Fig. 10 are required. Batteries are quite satisfactory, but for regular use a power pack is to be preferred.

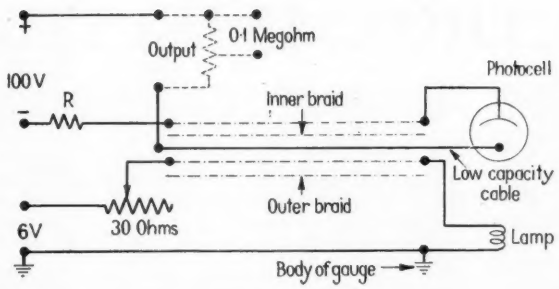


Fig. 10. Diagram of electrical connections

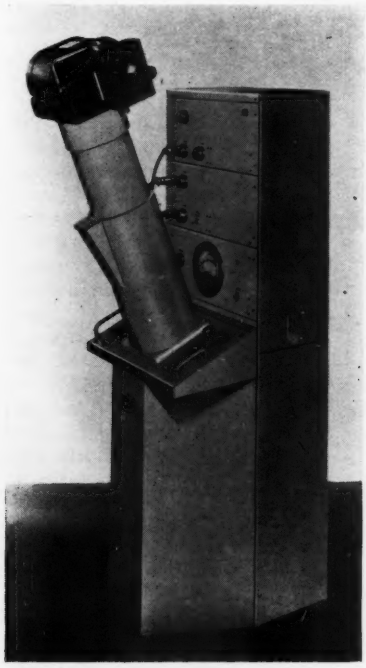


Fig. 11. C.A.V. oscillograph unit with 5 in. × 5 in. camera in position

Each gauge is supplied with 4 ft. of low-capacitance cable (10 $\mu\mu$ farad per foot) with two concentric braided screens which are used as conductors. The connections are as shown in the diagram and it will be seen that a connection in addition to those provided by the cable core and two braids must be made in the form of an earth wire to the body of the gauge.

The output is a current, the change in which is proportional to the change in shutter opening and for calibrating and setting up a current-measuring device may be connected across the points marked output in Fig. 10. When, for observation of transient phenomena on a cathode ray oscillograph screen, a gauge is to be connected to a voltage amplifier, the output is taken across a resistor connected across the output points. For the optimum frequency response the resistor should not normally exceed 100,000 ohms (c.f. output) and it should in no case exceed 2 megohms.

The C.A.V. Laboratories engine indicator equipments have been built as special tools, for use in investigation and development work in connection with fuel injection systems. The design of the apparatus has been kept simple and compact, and a layout employed which makes for easy operation by engine personnel.

Figs. 11 and 12 show an up-to-date unit with amplifier and time base mounted vertically, whilst the cathode ray tube and essential controls are mounted on an inclined panel in order that tracings of diagrams may be easily obtained if a photographic record is not required. Degree markings are spaced along the diagram at 10 deg. intervals with a 5 deg. mark on either side of "top dead centre".

The method of controlling the time base was fully described in our previous article. Briefly, it allows a diagram of the duration of a normal diesel fuel injection to be expanded, if desired, to the full width of the cathode ray tube without affecting the frequency of synchronization. It can also be operated either synchronized to an external pulse generated by the flywheel or allowed to run free and controlled only by the setting of the frequency oscillator. When synchronized it holds in step over a considerable speed range without adjustment.

The photographing of indicator diagrams has always been difficult even under the best conditions, for the reason that it is not possible to set the camera shutter with any precision to coincide with one revolution of the engine. Even if this were possible, the point at which the exposure starts can never be easily predetermined. On the indicator shown these difficulties have been overcome by making use of an electronic shutter* working directly on the cathode ray tube which provides the operator with complete control of the photographic cycle by the simple operation of a selector switch and push button. Three different exposures are available for a four-stroke engine :—

1. Compression and firing cycle.
2. Exhaust and intake cycle.
3. Cycle 1 *and* cycle 2.

The point at which exposure begins is determined by a trigger impulse generated at a fixed point with reference to crankshaft degrees. By phasing this point with reference to the particular part of the combustion cycle in which there is interest, either one or two revolutions may be photographed commencing at any one of the 360 degrees of either revolution.

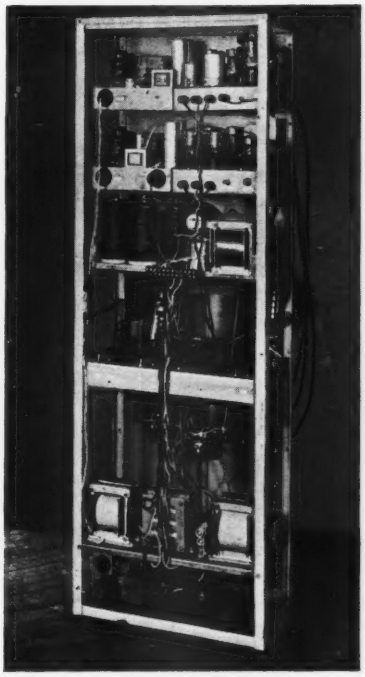


Fig. 12. Rear view of C.A.V. oscillograph unit

These gauges have created considerable interest among engine builders and others who have occasion to measure transient pressures and in consequence the firm have decided to make them available to other organizations and to supply the information or assistance required for their installation.

* A similar device was first seen working at the Thornton Research Centre by the C.A.V. engineers.

PETROL INJECTION

Special Bosch Equipment for Two-stroke Engines

INITIALLY Robert Bosch GMBH of Stuttgart concentrated their attention on the development of a reliable injection system for diesel engines and suitable to mass production. Experience thus gained was then applied to the problem of petrol injection for aircraft engines and, more recently, to develop petrol injection for the two-stroke engine. Research work over a period of some years was necessary to overcome all the difficulties. Equipment is now available that has been proved in practical use.

A positively controlled method of supplying and metering the fuel for each power stroke offers various advantages. Better facilities are provided for adjustment to running requirements. More efficient use of the fuel can be secured than is possible with a carburettor. In the case of the two-stroke engine, a number of special factors have to be considered, scavenging losses being one of the most important. Due to the superimposed functions in the two-stroke cylinder there is a certain loss of unburnt fuel during the scavenging process. These losses are reduced to the extent of 15 to 20 per cent. When direct injection is used, because in that system scavenging is done with fresh air only. Fuel is not injected until the scavenging process is nearly completed. It has been found unnecessary to wait until the exhaust port is closed before injecting the fuel. This is because the fuel during vaporization takes so much heat from the incoming air that the volume of air is reduced and the mixture does not pass out through the exhaust port.

In addition to the saving in fuel there is also a saving in lubricating oil. A special oil pump which forms a part of the injection unit is adjusted to minimum consumption and this quantity cannot be exceeded and the saving in oil is thus considerable. The actual economy claimed is from 30 to 50 per cent. A clean exhaust is another advantage of careful regulation of the oil supply.

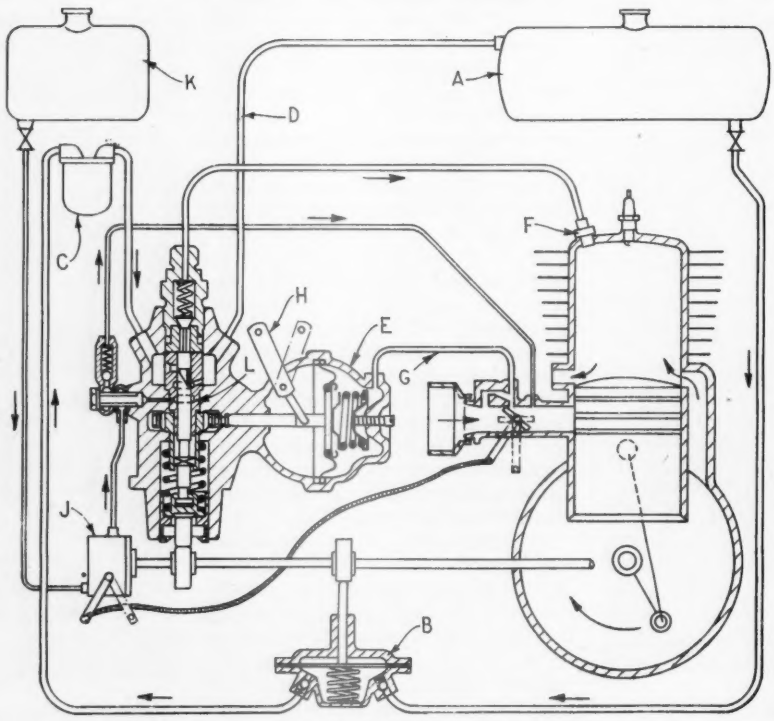
When driving conditions call for sudden accelerations and decelerations, the pneumatic governor of the injection pump reacts much more quickly to the changing demands than does a carburettor. The time interval between the moment when the accelerator pedal is moved until the altered injection mixture begins to be delivered is 0·02 sec. as compared with, at the best, 0·33 sec. for the corresponding reaction of a carburettor. Troublesome surging of the engine at full throttle is avoided with injection systems, because the fuel supply is positively limited by mechanical means as compared with the carburettor system, where the fuel supply depends on the air flow at all throttle openings.

Uneven running when idling is one of the outstanding disadvantages of a two-stroke engine for use in passenger cars. Direct injection gives much more favourable slow running conditions because even when the quantity of fuel delivered per stroke is small, the mixture is still maintained in a homogeneous state. With the carburettor, the functioning is inaccurate when the rate of delivery is low.

In view of the fact that enrichment beyond that obtained when the control is in the starting position is impossible, the manufacturers claim that starting is easier than with a carburettor. It is also suggested that an engine equipped with petrol injection is not so dependent for its good running on the octane number and voltatility of the fuel.

Two-stroke engines as a rule run at high speeds and have a comparatively small swept volume. This means that the quantity of fuel to be injected in each stroke is relatively small, but it must nevertheless always remain uniform. Piston-type injection pumps are used as a result of experience gained with diesel engines, where the same problems of accurate metering had to be solved at lower speeds. As in the case of the diesel engine, injection is effected at high pressure, to ensure atomization of the fuel against cylinder pressures, in contrast with some other systems in which fuel is sprayed at relatively low pressure into the eye of a supercharger.

Referring to the diagram it can be seen that fuel from the tank A is delivered by a diaphragm-type pump



A. Petrol tank
B. Fuel pump
C. Fuel filter
D. Fuel return
E. Pneumatic governor
F. Injection nozzle
G. Intake pressure pipe
H. Starting lever
J. Oil pump
K. Oil tank
L. Lubrication annulus

Bosch petrol injection system for two-stroke engines

B through a filter C to the injection pump. Fuel filtration is most important with petrol injection, for inaccuracies due to wear represent a comparatively large percentage error on the fuel delivery. Thorough development work has, it is claimed, provided the requisite high filtration efficiency in the Bosch cellular filter, which ensures adequate length of life for the injection system.

A deep annular groove in the pump body around the cylinder permits fuel vapour, liable to be formed at temperatures of 40 deg. C and over, to be returned with the over-flow of surplus fuel through the pipe D to the tank. On the downward movement of the piston fuel is drawn in and on the upward movement the rotational position of the helical cut-off face (controlled by the pneumatic governor E) determines the quantity of fuel delivered to the injection nozzle F.

A pipe G communicates induction pipe pressure to the diaphragm chamber of the pneumatic governor. Movement of the diaphragm, due to changes in induction pipe pressure, is converted into rotational movement of the piston and cut-off valve by the usual rack and pinion. An overriding control, actuated by lever H, is provided to give a rich mixture for starting.

Lubrication of the engine piston and cylinder as well as of the injection pump is furnished by a piston-type oil pump J, similar to the familiar Bosch oiler. Its output is varied according to the engine throttle opening, the throttle and oil pump controls being interconnected by a Bowden cable. Oil from tank K is supplied to an annular groove L in the injection pump cylinder, thus lubricating the piston and preventing fuel from finding its way into the cam chamber. A metered quantity of oil is also supplied through a pipe to the air intake. In cases where an engine is specifically built for fuel injection, as opposed to one converted from normal carburation, it is proposed that lubricant be delivered directly to the cylinder bore.

Experience gained with petrol injection nozzles for aero engines, as well as for diesel engines, was applied to the design of a suitable nozzle for the two-stroke engine. This takes the form of a small pintle-type nozzle and, contrary to the timing used in the case of diesel injection, is arranged to open at bottom dead centre. Turbulence induced by the new charge of air entering the cylinder is used to provide as homogeneous a mixture as possible in the region adjacent to the sparking plug.

At the high speeds, up to 5,000 r.p.m., at which the pump has to run, it was necessary to be particularly careful in the design and construction of the driving parts, and their lubrication must be well provided for. The manufacturers of this pump claim that such care has been given and it is unquestionably equal to all demands which may be made upon it.

To date, direct fuel injection equipment has proved to be too costly for installation on private cars. The prospect of its use for what would appear to be a simple, crankcase-compression engine for a low-priced car would seem to be rather remote. As in the case of the diesel-engined car, a very high annual mileage would need to be run in order that the possible economies in fuel and lubricating oil could offset the increased first cost of the vehicle.

RECENT PUBLICATIONS

Brief Reviews of Current Technical Books

Mechanics Applied to Vibrations and Balancing

By D. Laugharne Thornton, M.A., M.I.C.E., M.I.Mech.E., M.I.Mar.E.

London: CHAPMAN AND HALL, LTD., 37, Essex St., W.C.2. 1951. 6 × 9¼. 584 pp. Price 50s.

This is the second edition of this valuable work and embodies some additional references, some amendments, a new chapter, and a completely re-written General Survey. The new chapter covers the transmission of stress under conditions of rapidly applied loading. Examples of this occur, for example, in fortifications, ships, sometimes in materials employed for forming operations, and certainly when considering the possible effects of explosions. This chapter may have been influenced by the fact that the second world war has passed since the first edition of this book.

The new matter incorporated in the General Survey covers the transverse vibration of a turbine-wheel, vibration absorbers for engine crankshafts, springs, ship's hulls, buildings to withstand earthquake shocks, etc. This is a book for engineering designers, students of physics, and all interested in the subject of Theory of Machines. A fair standard in the calculus and analytical geometry is assumed by the author.

The author makes considerable use of Lagrange's equations for the solution of unbalanced dynamical forces and couples associated with engines and locomotives. He also shows the value of these, and Lord Rayleigh's theorem for the investigation of vibrations in structural systems having a finite number of degrees of freedom, and in the cases of complicated continuous systems.

After dealing with the balancing of engines and locomotives at considerable length, the theory of vibrations is discussed exhaustively. This is followed by Propogation of Stress in Elastic Materials, and in this chapter such matters as pile driving, reinforced concrete, water hammer, earthquakes, condensers, fuel systems of internal combustion engines, etc., are included. Under Beams and Plates are included continuous beams and structures, framed structures with rigid joints, membranes, beams of non-uniform cross-section, gyroscopic forces, the Principle of Least Action, and Hamilton's Partial Differential Equation. Rotating Shafts and Discs includes the whirling of shafts, strain energy, general theory of shafts, gearing, wheel and turbine blading, etc. The book remains a most authoritative work on its subject and is of importance to advanced students in engineering design.

Automotive Essentials

By Kay F. Kuns.

Milwaukee: THE BRUCE PUBLISHING COMPANY, obtainable from B. T. Batsford, Ltd., 5, N. Audley St., London. 1951. 6 × 9. 556 pp. Price 42s.

This book deals with the elementary essentials of automobile construction and is intended for students and all those who are not trained automobile engineers. It concerns automobiles made in the United States only, and it is important to understand the reason for the book.

The motoring public in the United States use their motor cars merely as beasts of burden. They know nothing about the mechanism and generally do not wish to. It is true to state that most of the drivers have never looked under the bonnet of their cars. Service stations in the populated districts are not far between and the user just drops in on them for every job, however small. There is no publication in the States such as *The Autocar* for the general motorist.

The author of this book has realized, however, that there must be a great number of car owners who are mechanically minded and who wish to know the essential parts of their beasts of burden and how they work. In any case, budding service station mechanics will be interested. This book covers the subject from this angle, and may be read by any motor car owner. It deals, of course, only with American vehicles, and because of this, it could be distinctly interesting to many people in this country, including those mechanics who are sometimes called upon to service American cars.

In spite of the fact that it deals only with American cars, the book is most interesting and is very well written. It starts from the very first of first principles. For example, there is a chapter on Names and Uses of Engine Parts, the chapter on Lubricating the Engine begins with a sub-heading on the "Need of Lubrication," and similarly on Cooling the "Need of Cooling," is first explained.

The illustrations form an important part of the book. They are very numerous and

include exploded diagrams and valuable sectional drawings of almost every automobile part. The whole of the motor car is covered in addition to engine details, including such items as automatic transmission, power transmissions, motor cycles and scooters, starting motors and generators, batteries, wiring, brakes, steering gears, etc., etc. Each chapter concludes with a number of Questions and also Research Questions and Problems, such as "Do you know of any car manufacturer who advertises his engine brake horsepower," "Give the steps for the Ross Gear," "To which type of gears does the Gemmer gear belong," "What will be the result of allowing a battery which is discharged to be chilled below 32 deg. F", "What cars, if any, use a yoked type of connecting rod?"

This book can be recommended to anyone interested in the mechanism of motor cars. It has its counterparts in several publications in this country, but it is doubtful if there is one that covers quite the same ground.

Trader Handbook, 1951

London: TRADER PUBLISHING CO., LTD., Dorset House, Stamford Street, S.E.1. 1951. 458 pp. 5½ × 8¼. Price 12s. 6d.

This, the 45th Edition, embodies all necessary amendments to bring it up to date. It is a handbook of value to both the buying and service departments of motor, motor cycle and cycle dealers. It provides a brief legal guide explaining the principal points and exposing many current errors. The technical section gives dimensional details of every British motor car, commercial vehicle and motor cycle manufactured from 1939 to 1950, some particulars of American models, and many diagrams. One section gives licensing information and index marks. Another section, entitled Transport Service Equipment, lists in alphabetical order, makers of everything from goggles to gear cutting machines. The Buyers' Guide, which follows this, is still more comprehensive and occupies more than a hundred pages. Finally, there are the Proprietary Names and Trade Addresses.

New and additional matter includes a table showing the names of proprietary brands of lubricating oil, corresponding to SAE numbers for engine and gear oils. The Index of Registration Numbers reintroduces this year a list of the Licensing Authorities concerned. New material for the cycle dealer and repairer has also been included, particularly in connection with wheel building.

The pages are interleaved with manufacturers' announcements, which are not out of place in a book of this type, and the Handbook maintains its reputation as an up-to-date and reliable reference work for the motor and cycle trades.

Kraftfahrzeug-Bremsen (Motor Vehicle Brakes)

By A. Pleines.

Stuttgart: BERLINER UNION G.M.B.H., 12, Hospitalstrasse, Stuttgart, N. Germany. 1951. 6 × 9. 225 pp.. Price DM40.

Dealing exclusively with motor vehicle brakes and braking systems, this German book will be of value to car and brake manufacturers and designers. It embodies the conclusions of twenty years' specialized experience by the author. Starting with the first brake, for a hand cart, it refers to the braking systems used on early motor cars at the close of the nineteenth century and then discusses the theoretical side of the subject and details of design. Every type of brake is considered, including hydraulic, compressed air, vacuum and servo-assisted systems.

One chapter is devoted to operating details such as brake levers, pistol grips, and controls, and there seem few omissions from early days right up to the present. Obviously most of the systems discussed and illustrated are German, but one sees such names as Girling, Bendix, Dewandre and others. German practice will certainly be of interest to brake designers in this country. The illustrations are also very valuable, depicting everything from component parts to complete systems, for vehicles ranging from small cars to large lorries and trailers. The author also has many remarks to make concerning future tendencies in brake design and it is interesting to compare these in the light of recent experience under present, and possible future, traffic conditions in our own country.

Production of Motor Vehicles

By Henry M. Cunningham and William F. Sherman.

U.S.A.: MCGRAW-HILL PUBLISHING CO., LTD., Aldwych House, London, W.C.2. 1951. 6 × 9. 169 pp. Price 30s.

This must be the first book to deal with the planning that goes into the production of a new type of model from the first design stage to the completed prototype. It comes from the U.S.A. and should, perhaps, be of greater interest on that account, because the production rate of some automobile factories there is so much greater than our own. The book is semi-technical, and might well be of interest to anyone connected with the production of cars or commercial vehicles.

The chapter titles give an accurate lead to the contents of the book, namely, the Inception of the New Model, Administration and Technical Problems Resulting from Proposed Model Changes, Preliminary Planning for New-Model Production, Completion of the Production Plan, Controlling the Production Plan, Control of Quality by Inspection, Cost Control, and finally Distribution Techniques. There is therefore something in this book for production executives as well as automobile sales departments. Every automobile production engineer will learn from it, and most production engineers engaged on making automobile parts will also read with interest.

The book was originally written for the Industrial College of the Armed Forces in the U.S.A. and to meet a need of several branches of the National Military Establishment. The work was undertaken without cost to the government as a public service by the automobile industry, and all Members of the Automobile Manufacturers Association have assisted. Two years was spent in gaining first hand knowledge of the basic principles underlying practices and techniques peculiar to the automotive industry. No attempt is made to present descriptions of the detailed operations of automobile production.

The many charts are valuable, they include design work order, draughtsman's design log for mechanical change, building order for experimental unit, test reports, lead-time charts for purchased parts, daily schedule for delivery of materials, assembly timetable, etc., etc., right through to dealer new-car projection sheets and order forms. The information is not particularly detailed, and is not intended to be, but much of the material may be applied to many large scale production ventures outside the automobile industry.

Automotive Engines Maintenance and Repair

By Erbest Venk, M.A., Voc.Ed.

Chicago: AMERICAN TECHNICAL SOCIETY. Obtainable from The Technical Press, Ltd., Gloucester Road, Kingston Hill, Surrey. 1951. 5½ × 8¼. 381 pp. Price 36s.

This book on motor car repair and maintenance is written by an American engineer, and deals with American practice. It covers engines and parts from American cars. The author is an instructor in automobile and aviation mechanics at the Fordson High School, and the book is written with the training of automobile mechanics in mind. It covers the subject as comprehensively as could be done in a single volume and includes almost everything from engine nomenclature and the development of the internal combustion engine to the assembly of the modern car.

A valuable chapter for the young mechanic is that dealing with cleaning and measuring, since the cleaning of an engine is a necessity when repair or rebuilding is undertaken. An appreciation of precision instruments is therefore essential for the engine specialist. These two items show that the author is alive to detail and the book covers the subject in a new, but practical, manner.

Chapters describe and illustrate Dismantling, Cylinder Heads, Cylinder Blocks, Crankshafts, Pistons, Valves, Engine Lubrication and Cooling, Assembly, and Re-machining. This is a subject that only specialist engine mechanics to-day understand and the ability to re-machine engine parts, instead of merely to replace them, divides the craftsman from the usual run of motor mechanics.

Foremanship

By F. J. Burns Morton.

London: CHAPMAN & HALL, LTD., 37, Essex Street, W.C.2. 1951. 5½ × 8½. 316 pp. Price 25s.

This book should not be confused with "The New Foremanship" or "Introduction to Foremanship," both of which have been written by the same author. It is a book for study by foremen, future foremen, and business executives who have control of or who work with foremen. In his foreword the author says "A course of lectures, and still less a textbook, cannot alone produce the first-class foreman. The character and personality of the foreman, his general education and individual adaptability are of primary importance. With capable men, formal instruction with the aid of a textbook will assume its proper function of providing the tools of management to be used as circumstances demand."

Although many foremen are born, and not made, the author rightly considers that the art of foremanship must be supplemented by science and theory. In this book he shows ways of lessening the friction associated with a foreman's duties.

The foreman's place in connection with operation, production, quality and cost control, have each their respective chapters after the first section of the book has been devoted to the three subheadings of Direction-Supervision, Organization, and Management. The third section, Personnel, covers training, incentives, labour management, working conditions, and leadership. These may appear to be the same titles that are found in any book on this subject or on works management generally. While much of the material is similar, the clearness of exposition makes the book very readable.

AIR CONDITIONING*

Some Factors in the Design of Equipment for Passenger Vehicles

By F. W. Duncombe+

(*continued from page* 309)

Requirements and applications

It is now necessary to consider the Thermal requirements for a passenger vehicle air conditioning installation. It has already been stated that the human body gives up a certain amount of heat and this is in the region of 400 B.T.U. per hour, and is made up by approximately 300 B.T.U. per hour sensible heat and 100 B.T.U. per hour latent heat. It is also interesting to note that under normal seated conditions, the average individual gives off by evaporation and breathing something like 0.1 lb. of water vapour per hour.

In a 30-seater coach which is full of passengers, there will be something like 10,000-12,000 B.T.U. per hour added to the air in the coach. This figure is quite appreciable and it has been suggested that when considering the design of a passenger vehicle coach installation, the supply of heat from a full load of passengers will account for the heat which is lost through the body panel and windows of the vehicle, provided that the T.D. between the inside and outside temperatures is not excessive. It has also been suggested that for an adequate supply of oxygen, and to enable the concentration of carbon-dioxide to be adequately suppressed, the normal individual requires something like 20-30 cu. ft. per minute.

* A paper presented to the Institute of Road Transport Engineers Ltd.

+ Development Engineer, The Clayton Dewandre Company Ltd.

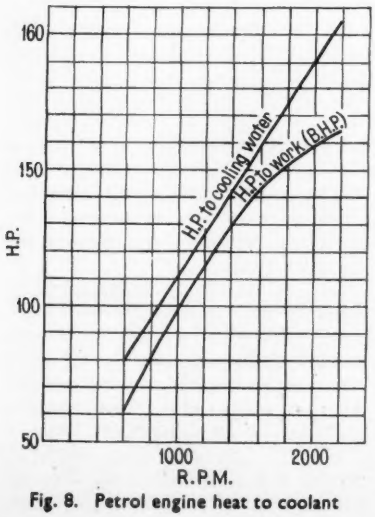


Fig. 8. Petrol engine heat to coolant

The stage is now reached where it is possible to obtain at least some idea of the size of equipment which is required to meet the operating conditions—whether they be for heating or for cooling. Assuming that the design capacity for the heating equipment required is based upon the outside conditions prevailing, it is possible then to calculate the requirements of the inside heating equipment.

Usually one designs for the worst conditions that are likely to be met, that is to say, for heating, one would use the winter temperature and humidity for the zone of operation, and base the requirements of the internal heater on the data already given. A formula which would give a reasonable assessment of the duty required would be :

$$\frac{CFM \times 60\{\text{Indoor enthalpy} - \text{Outdoor enthalpy}\}}{\text{Air density (cu. ft. per lb.)}} = \text{B.T.U. per hour heater capacity required.}$$

Should the equipment be required for cooling purposes, then it is necessary to determine the required cooling capacity of the refrigeration equipment. This can be assessed from a like formula in a similar manner :

$$\frac{CFM\{\text{Outdoor enthalpy} - \text{Indoor enthalpy}\}}{\text{Air density (cu. ft. per lb.)} \times 200} = \text{Tons of refrigeration capacity.}$$

This is based on the assumption that one ton of refrigeration is equivalent to the extraction of approximately 12,000 B.T.U. per hour, or 200 B.T.U. per minute.

It is stressed that the heating and cooling equipment capacities as suggested by these formulae are for the respective outdoor air load only.

It is obvious that in a paper of this description, one cannot give all the technical data for a particular installation. So much will depend upon the design of the vehicle, where it is to be operated and its service conditions. In certain installations, where cheapness is called for, and local service conditions are encountered, a recirculated type of unit may be used. Also to be considered for similar types of service is the double deck vehicle, and in this case it is desirable to heat or condition both the upper and lower saloons. Finally, there is the luxury type of vehicle which is engaged on long distance runs, when a more elaborate system would appear to be necessary. As a rough guide, the following ratings for the capacities of the heating units required are given :

(a) A 26-30-seater public service vehicle would need a unit capable of about 15,000 B.T.U. per hour. This would provide an approximate temperature rise of 20-25 deg. F.—that is to say, when all the air is being recirculated and provision is not made for the admittance of fresh air.

(b) In the case of a double deck vehicle of—say—40-seater capacity and where provision is made for the admittance of fresh air, it is suggested that the heater capacity should be in the region of 35-40,000 B.T.U. per hour. This would give something like a 25-30 deg. F. rise in air temperature.

(c) For a coach operating in temperatures of about 20 deg. F., a dissipation of about 60,000 B.T.U. per hour would give reasonable comfort conditions with well insulated bodywork.

Available source of heat

For the purpose of this paper, the source of heat available for the heating of the vehicle will be considered as that which is available in the engine cooling water. Other alternatives and possibilities could be considered, but it is not proposed to discuss them. Consider now the actual amount of heat which is available from the engine cooling water. Experiments have proved that for the normal petrol

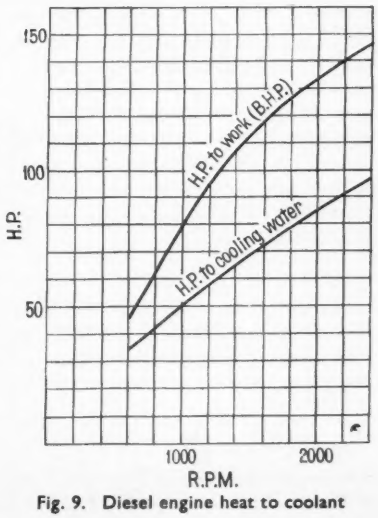


Fig. 9. Diesel engine heat to coolant

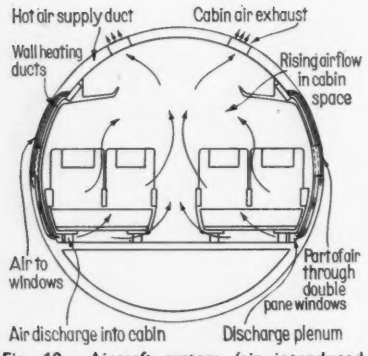

Hot air supply duct
Cabin air exhaust
Wall heating ducts
Rising air flow in cabin space
Air to windows
Part of air through double pane windows
Air discharge into cabin
Discharge plenum

Fig. 10. Aircraft system (air introduced under seats)

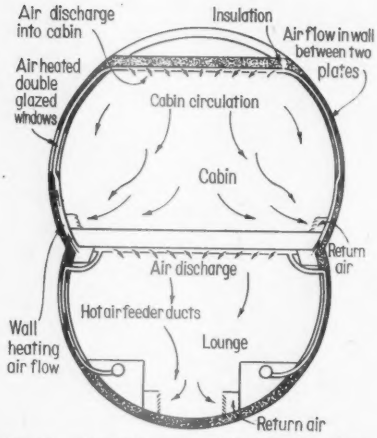


Fig. 11. Aircraft system (air introduced at the cabin roof)

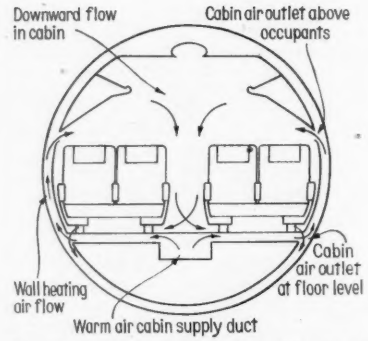

Downward flow in cabin
Cabin air outlet above occupants
Cabin air outlet at floor level
Wall heating air flow
Warm air cabin supply duct

Fig. 12. Aircraft system (air introduced from cabin walls)

engine the heat to the cooling water averages 1·2 times that of the b.h.p. developed (see Fig. 8). For a diesel engine, the heat to cooling water is approximately 0·75 of the b.h.p. developed. (See Fig. 9.)

Analysing this further, take the diesel engine as being the worse case; for an engine which is developing 100 b.h.p., according to the above figure the heat to cooling water would be 75 h.p. Transposing this amount of heat into terms that are more familiar, the amount of heat available is 75×2540—which equals 192,000 B.T.U. per hour, and carrying this a stage further—

$$\frac{192{,}000}{3{,}411} = 56 \textit{ kilowatts}$$

These figures show that for most of the operating conditions likely to be encountered, there is an adequate source of heat available.

The sizes of unit which are in most demand today are for capacities of from 15–30,000 B.T.U. per hour dissipation, at a temperature difference of 120 deg. F. This temperature difference figure would be the air inlet temperature to the mean water temperature.

Taking the unit having 30,000 B.T.U. per hour dissipation for the foregoing T.D., revert for a moment to earlier remarks regarding temperature difference—it is appropriate to give examples of the change in performance for alternative temperature differences. If the unit is required to operate at—say—a temperature difference of 80 deg. F., then the new performance figure will be :

$$\frac{30{,}000}{120} \times 80$$

which equals 20,000 B.T.U. per hour. On the other hand, if the temperature difference is greater and is—say—140 deg. F., the dissipation achieved will be :

$$\frac{30{,}000}{120} \times 140$$

which equals 35,000 B.T.U. per hour.

These figures readily show that for a given heat extraction required, the number of units required to heat adequately the vehicle depends entirely upon the operating temperature difference conditions. Once the capacity of the heating unit is known, it is possible to derive other useful information. The average unit delivers something in the range of 200–400 cu. ft. of air per minute. For a given dissipation and a known air flow, the air temperature rise across the heater can be calculated readily from the following formula :

$$\textit{Air rise (deg. F.)} = \frac{\textit{B.T.U. per hour}}{1.1 \times \textit{CFM}}$$

The formula is based upon the assumption that the air is measured at 60 deg. F.—thus it will be seen that for a unit having a capacity of 30,000 B.T.U. per hour and an air flow of 250 cu. ft. per minute, the air temperature rise would be 100 deg. F. This air rise is naturally dependent upon the required dissipation at the specified temperature difference.

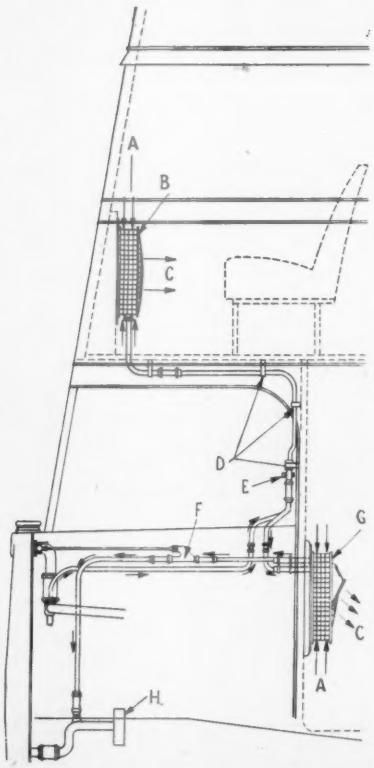


Fig. 13. Double-deck bus recirculating installation

Current developments and installations

In the opinion of the writer, the existing installations and equipment will hold the field at least as far as this country is concerned for some considerable time to come. It is, however, advisable to present for consideration, ideas which could be applied to the air conditioning of passenger vehicles. Considering the case of vehicles which are operating in conditions of extreme cold, it might be advisable to consider heating the wall of the vehicle as is sometimes used in aircraft installations. The illustrations Figs. 10, 11 and 12, give some idea of what is involved and it is interesting to consider the functioning of the various systems.

In all cases, the designer has endeavoured to reduce the thermal exchange between the inner and outer skins and the wall by passing heated air between the inner and outer skins used in the construction. Also, double wall windows are incorporated and this very much facilitates de-icing and tends to prevent condensation by the passage of the heated air between the two glass panes.

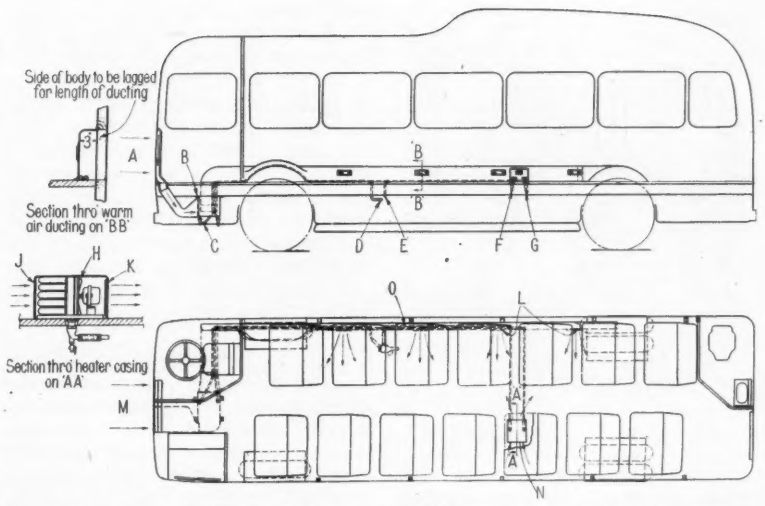
In Fig. 10, the air is delivered by ducts to the space between the wall, passes down the wall through the window units and out into the main saloon under the seats of the passengers This air will, in due course, rise and is drawn out by suitable extractors.

Fig. 11 indicates what might be termed a double-deck installation and here, again, air is delivered by ducts

to the appropriate part of the body skin where it passes over the windows —but in this case, it is in the reverse direction—i.e., it is passing upwards and is admitted into the respective saloons through the roofing which consists of very finely perforated sheeting. The return air is drawn into suitable ducts in the lower part of each saloon.

In the third installation—Fig. 12—a variation of that shown in Fig. 11 is used. In this particular application, warm air is forced through a central ducting underneath the floor. From this ducting, connections are made to the outer skin, through which the air flows to outlets which are situated at higher level. At various points at floor level return apertures are provided so that in the main the flow of the air is downwards in a similar manner to that illustrated previously.

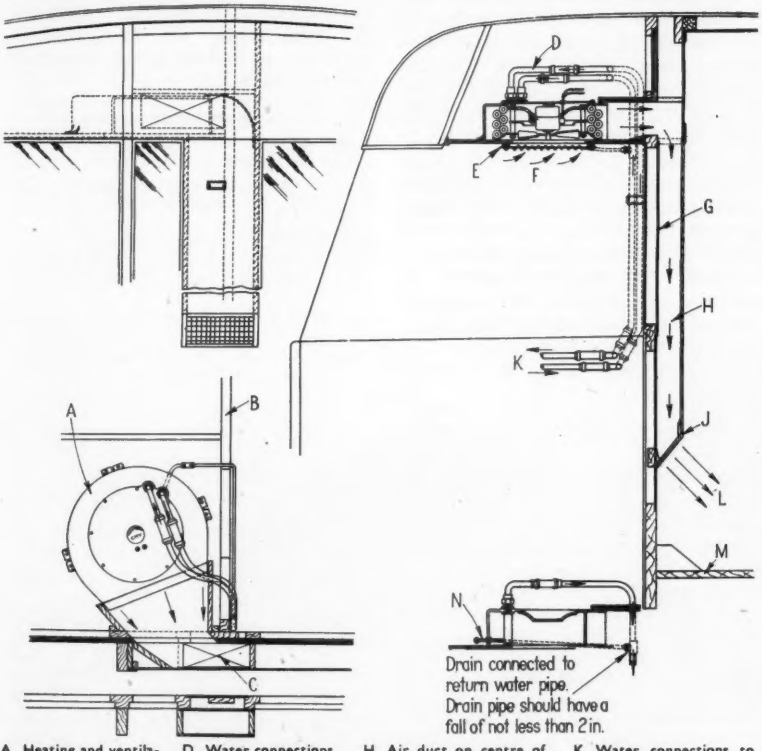
Certain independent authorities have, within recent years, carried out investigations into alternative methods of providing comfort for the passengers, and attention is drawn to an experimental unit which is running on the roads at present. This unit was instigated by the Motor Industry Research Association and the principles involved are two-fold—i.e. to cool the engine and at the same time to heat the passengers.



A. Air inlet.
B. V.25 1A heater unit.
C. Lagged pipes under floor.
D. Heater water return.
E. Heater water delivery.
F. Elbow drain assembly.
G. Elbow air vent assembly.
H. V12/2 heater unit.
J. Gauze at re-circulating air inlet.
K. Warm air outlet grille.
L. Warm air delivery.
M. Fresh air inlet at waist rail height.
N. V12/2 re-circulating air only.

Fig. 15. Single-deck bus underfloor installation

On this unit, the heat exchangers are situated in the canopy above the driver. The main feature of the system is that as the units are placed horizontally, vertical air flow can be employed. By this method, assistance is obtained by convective flow. This flow is created by the thermal temperature difference between the inlet air and the temperature rise of the air as it passes through the heat exchanger. When the vehicle is in motion, this convective flow is aided very considerably by the speed of the vehicle. The heated air after leaving the heater is fed by suitable ducting to the saloons. For summer conditions, suitable dampers which are incorporated in the system allow the heated air to be diverted to the outside atmosphere.



A. Heating and ventilating unit.
B. Cab side.
C. Air ducting.
D. Water connections.
E. Inlet grille.
F. Fresh air inlet.
G. Lining panel for air duct.
H. Air duct on centre of bulkhead, approx. 30 sq. in. internal area.
J. Honeycomb outlet grille
K. Water connections to engine.
L. Warm air outlet.
M. Floor level.
N. Drain pipe.

Fig. 14. Single-deck bus fresh air installation

A second installation which is worthy of mention is the one developed by the London Transport Executive. In this instance a heat exchanger is situated under the staircase at the rear of the vehicle. This heat exchanger is used for cooling the engine and at the same time to heat the upper and lower saloons.

Arrangements are provided in this installation for reversing the air flow. For summer conditions the fan acts as an exhauster—the air is drawn from outside, passed through the saloons and is then blown through the heat exchanger. In this cycle of operation, fresh, relatively cool, air is continually being supplied to the passengers, and at the same time the air is being used for cooling the engine. For winter operation, the system is reversed, so that air is drawn initially from the outside of the vehicle, is heated on its way through the heat exchanger and is then distributed through suitable ducting into the saloons—so providing heated conditions for the passengers.

More orthodox installations which are in everyday use are illustrated in Figs. 13, 14 and 15. Fig. 13 is a

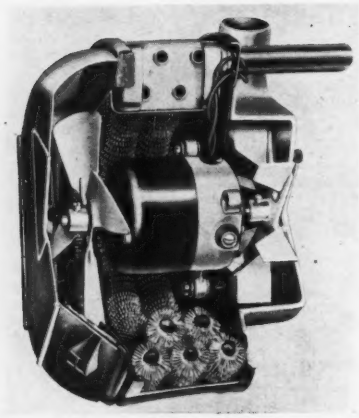


Fig. 16. Typical cab heating and demisting unit

typical installation for a double-deck type of vehicle where a recirculated air system is considered to be satisfactory. Fig. 14 shows an installation where the heater is fitted into the canopy above the driver. Fresh air is admitted to the heater, is warmed and then discharged through a suitable ducting into the interior of the coach. Fig. 15 indicates a typical installation where the heater is situated under the floor and the air is discharged from suitable outlets in ducting attached to the floor of the vehicle. To augment the supply of heat, a smaller type of heater is situated under the floor; this latter works on the recirculated air system and assists in a relative quick warm-up of the coach.

Last, but not least, the comfort of the driver must be considered. This is really a subject apart from the providing of necessary comfort for the fare-paying passenger. At all times the driver should be warm and reasonably comfortable, and what is more essential, his windscreen should be clear of mist and, when occasion demands, should be de-iced.

These factors are obviously essential for the safety and well being of all road users. Regarding the early remarks relating to humidity, it was shown that by raising the temperature of the air, it was possible to make the air capable of absorbing more moisture and so preventing precipitation. For this duty it has been proved that a unit capable of—say—dissipating approximately 5,000 B.T.U. per hour would be satisfactory for all normal conditions. Illustrated in Fig. 16 is a typical cab heating and demisting unit and shown in Fig. 17 is an indication of how such a unit can be installed.

Conclusion

As the title of this paper implies, the author has tried to cover some features which affect the heat exchanger design, the conditions which have to be considered to give the passenger reasonable comfort, and to give an indication of where present systems might be improved. No reference has been made to the various types of controls which would be necessary to give automatic or semi-automatic control of the comfort condition required. Some of the instruments obviously required would be—temperature thermostats, humidity controls and devices for controlling the amount of fresh air admitted, relative to the conditions and temperature desired inside the vehicle.

The satisfactory design of duct work is also an essential for the satisfactory functioning of an installation. The ducting layout needs very careful thought, and it is rather surprising how the area of the ducting, and the length, affect the air flow. The openings in the ducting for the admittance of air into the vehicle also require very close investigation, particularly with regard to even distribution, and velocity of the air from the respective openings.

Finally, the design of suitable air conditioning systems for passenger vehicles will most probably be influenced by what has been done successfully in the past rather than by the

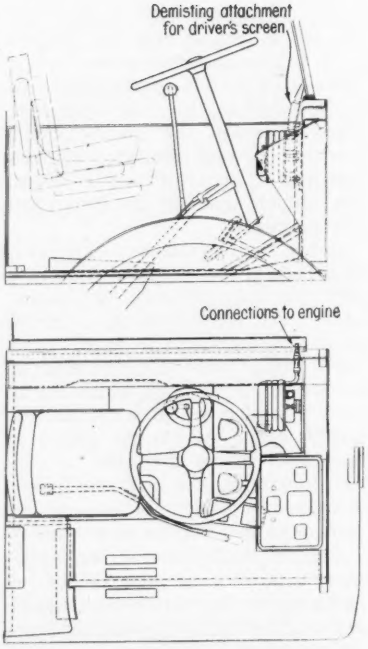


Fig. 17. Cab heater and demister installation

results of a direct investigation into the factors outlined in this paper. It is, however, true to say that the influence of the factors mentioned will be felt, probably indirectly, but all the same they will obviously determine the final results achieved—whether they are covered by actual design or rule of thumb.

Acknowledgments

Acknowledgements are made for the use of the following illustrations.

Fig. 5 } "Air Conservation Engineering."
Fig. 6 } W. B. Connor Engineering Corp.

Fig. 8 } S.A.E. meeting, January, 1950—
Fig. 9 } "Air Conditioning of Automotive
Fig. 10 } Vehicles," by A. T. Browne.

Fig. 13 } S.A.E. meeting, October, 1946—
Fig. 14 } "Fresh Air Automotive Heating—
Fig. 15 } Influence of Aviation Industry,"
by L. A. Rodert.

CARAVAN DESIGN

THE design and development committee of the trade division of the National Caravan Council have approved the following recommendations: All caravans should be fitted with jockey wheels, and those vans in excess of 1 cwt. on the tow bar should have screw lift or hydraulic type jockey wheels. The committee, after due consideration of the information supplied by coupling manufacturers and representatives of the caravan trade, concluded that whilst it was desirable for there to be one standard hitch throughout the trade, such a proposal was impracticable for the present. In view however, of the similarity of various hitches at present in use and in order to avoid the possibility of mistakes by caravan towers using the wrong car attachment, the committee recommend that coupling manufacturers should consider stamping their names on both the inner and outer portions of their couplings.

The committee were also instructed to investigate the question of the use of safety chains to prevent the caravan and towing vehicle parting company. The committee stated that on the evidence available it did not feel that there was any justification for recommending the use of safety chains when towing caravans.

Other matters under consideration by this committee are the use of Weyroc for caravan flooring, and hardboard problems. A recommendation on the use of Weyroc will be made shortly, and the committee hope to have discussions with the appropriate Association with a view to obtaining an improvement in the present standard of hardboard used by the trade. (1965)

STANDARDIZED FIXTURES

The Purefoy range of units for Tooling

TWO important factors that must be considered when the tooling is being arranged for new production are the cost of the jigs and fixtures and the time necessary to make them. When, as frequently occurs, every jig or fixture is specially made in the tool room, the cost is relatively high and the time taken may be considerable. For many applications the costs of jigs and fixtures can be greatly reduced and the time that must elapse before production starts can be greatly reduced if advantage is taken of the Purefoy system of unit tooling.

Some years ago J. B. Purefoy Unit Tooling Ltd., Cobham, Surrey, investigated the possibilities of producing a range of standard parts for use in jigs and fixtures. A wide and detailed analysis was made of the individual items that go to make up jigs and fixtures in general. From this analysis it was possible to draw up a list of items that recur constantly, and which were therefore suitable for standardization. These items are produced in such quantities as to permit of manufacture to close limits of accuracy at relatively low unit cost. They are therefore obtainable at prices lower than for similar articles produced either singly or in small quantities. Secondly, since every item in the Purefoy range can normally be supplied from stock, the time lag between the approval of a design and the start of production can be greatly reduced. There is also an appreciable saving in drawing office work.

Chief items

It is not possible to discuss in detail all the many parts that are available, but brief descriptions of the chief items in the range will convey some idea of the potentialities of the system.

Standard sections. There are several different shapes of sections, including "H" sections, square and rectangular hollow blocks and "L", "U" and "V" sections. Every type of section is made in close grained grey cast iron. Hollow blocks are machined on all outside faces and all other sections are machined all over. For many applications a piece of appropriate section will form a complete jig body without any machining other than cutting to length.

Base plates. These are also made from grey cast iron and are machined complete with holding down lugs, tenon slot and datum faces. Many kinds of machining and inspection fixtures can be built on these base plates.

Circular adaptors. These are available either as plain circular blanks or else ready recessed and drilled to fit directly on to the spindle of a capstan lathe. A second type has two different diameters. It provides the basic element for many turning and rotary milling fixtures.

Jig plates. The jig plates are ground on both faces and have four reamed holes so that by the addition of four feet bolts and four feet nuts the whole framework of a plate jig is immediately available. The feet bolts and nuts are of hardened and ground steel.

Hand and thumb nuts. Hand nuts are made in cast iron and thumb nuts in nickel plated steel. They are available with either Whitworth or B.S.F. threads. Each kind can be supplied in a quick-release type that needs only a quarter-turn for locking or releasing.

Clamping devices. A wide range of parts is available for fabricating clamping devices. It includes plain and slotted clamps, clamp springs, "C" washers both plain and captive, spherical washers and seatings, heel pins and domed nuts.

Typical applications

Four of the many applications for which Purefoy unit tooling has been employed are shown in the accompanying illustrations. A latch type drill jig is shown in Fig. 1. The body of the jig is made from standard "U" section. The component is held in position by a clamp plate that fits very freely on the latch pin and is locked in position by a screw in the latch. In the jig as illustrated, a standard quarter-turn thumb screw is used for locking the latch, but a standard swing bolt could be used equally well.

An efficient turning fixture that is completely free from swarf traps is shown in Fig. 2. It comprises a standard capstan back plate fitted with two vee jaws which are centralized and clamped by means of a cone-end screw. The vee jaws are spring-loaded so that they open automatically when the coned screw is withdrawn. The operations performed with this fixture are :— bore and face large boss, face arm of rod and face small boss.

A jig designed for drilling, counterboring and spot-facing three holes in both right- and left-hand components is illustrated in Fig. 3. The hole centre of the components are too close to allow the normal method of using slip bushes to be employed. Because of the close centres a swing latch plate is fitted with bushes to suit the tapping size hole. It is retained by a quarter-turn thumb screw. After the tapping size hole is drilled the latch plate is released and swings clear to allow the counterboring

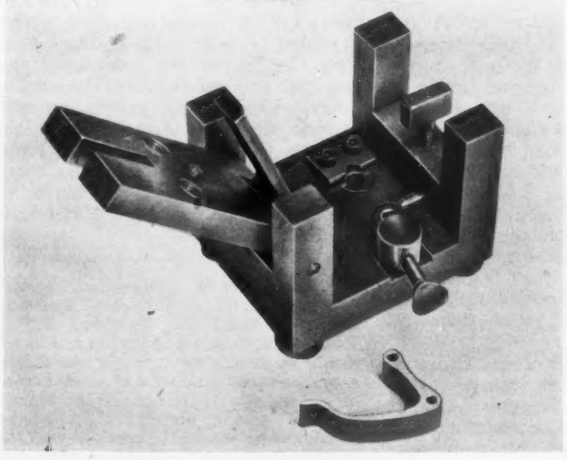


Fig. 1. An efficient latch type drill jig built from Purefoy standard units

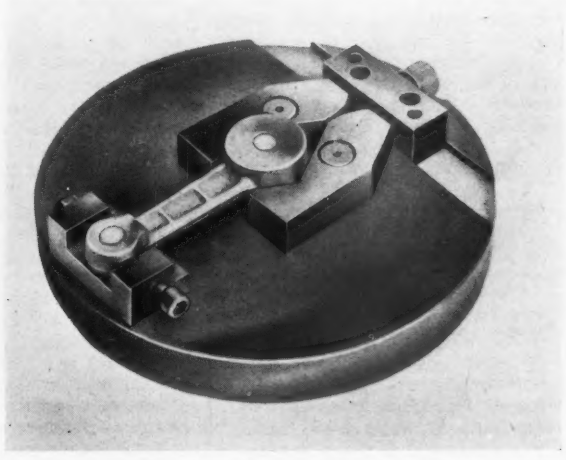


Fig. 2. A turning fixture built on a Purefoy standard capstan back plate

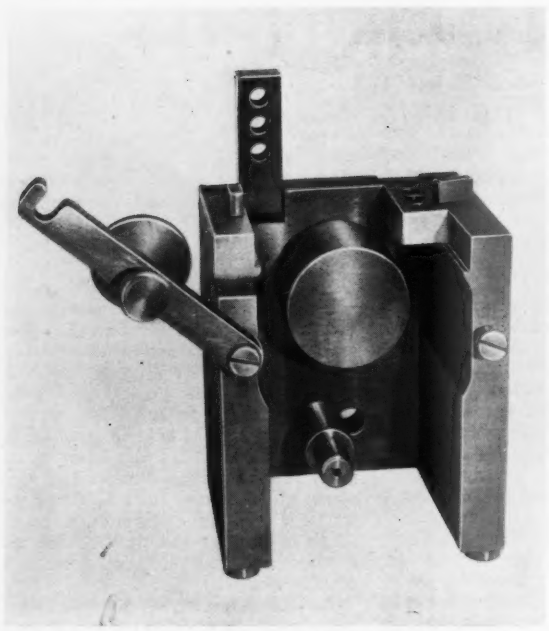


Fig. 3. A drilling fixture that will accommodate both right and left hand components

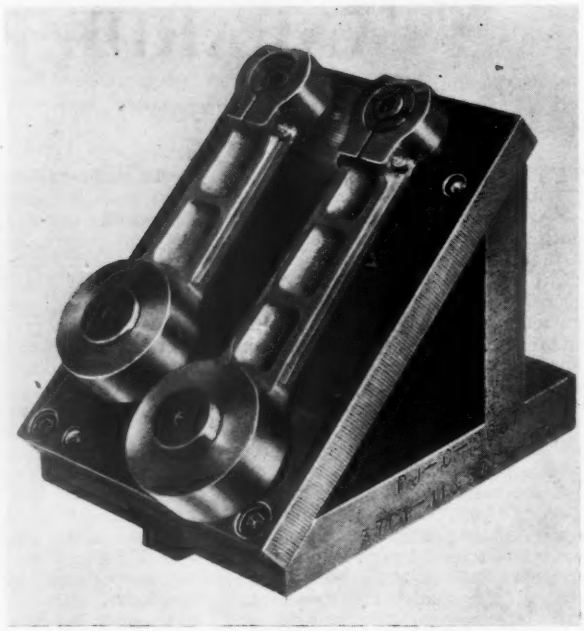


Fig. 4. A profile milling fixture built mainly from Purefoy standard units

and spot-facing operations to be carried out.

For opposite hand components, the small locator is changed to the other hole in the body and the other swing bush plate is used. It should be noted that the jig body, feed buttons, shoulder screws, swing latch, clamping pad and quarter-turn thumb screws are all Purefoy standard parts. An interesting profile milling fixture is shown in Fig. 4. For this fixture a standard unequal "T" section has been machined to suit the angular setting. Standard tenons are fitted in the base. Each component is positioned by a spigot in the large end and an expanding spigot in the small boss.

SPUR GEAR PUMPS

THE author of "*Spur Gear Rotary Pump Design*," T. F. Fitzgibbon, in *Product Engng.*, January and March, 1951, discusses the applications and the basic factors affecting the design of spur-gear pumps.

Applications for gear pumps are limited only by the maximum sustaining pressure of 1,500 lb. per sq. in. and by their unsuitability for variable volume operation. Simplicity of design results in easy maintenance and a small stock of replacement parts.

Factors affecting the design are: operating pressure, peak pressure, output, speed, type of fluid, operating temperature. In the preliminary stage a rough estimate is made of the pitch, diameter and the width of the gears, assuming that the tooth space area and the tooth area are equal. The possibility of getting sufficiently large journals is checked next, taking into account the hydraulic load only. From the lay-out the tooth space area and the length of tooth contact along the line of action can now be measured. A high pressure angle pyramid form tooth is customarily used with a pressure angle of 25–28 deg. Bearing loads can be conveniently analysed graphically. They consist of the hydraulic or side load, the tangential force and the separating force. The resulting load vector is inclined by about 20 deg. to the horizontal centre line on the inlet side. Journal and needle bearings are commonly employed; the former are cheaper and more reliable than the latter, but require more diametral journal clearance.

Leakage is a result of end clearance (radial and transverse leakage) or of diametral clearance. In general, leakage varies as the cube of clearance and can constitute a serious loss unless minimum clearances and good alignment are ensured. The oil circulating through the bushings may be drained and returned to the tank, or it may be redirected to the suction side of the pump through a series of holes in the body. The housing design can consist of two or three plates shrouding the gears. It is important that their coefficient of thermal expansion should be similar to that of the gear material.

To prevent leakage between the end plates and the shroud a shallow groove is milled in both faces of the centre plate. It follows the contour of the gear bore and channels into the suction cavity. Alternatively, gaskets are employed of thin, little compressible material, such as aluminium. Fluid trapped in the bottom of the tooth spaces must be given an outlet to ensure quiet performance. (*M.I.R.A. Abstract* 5348).

WASTE PAPER

EVERY industry is vitally dependent on paper in some form or other. In 1950 600,000 tons were used solely in packaging. Increased exports mean increased demands. Save all waste paper. The Waste Paper Recovery Association, 52, Mount Street, London, W.1 will be glad to furnish addresses of local merchants.

A new series of information sheets is being prepared by **Acheson Colloids Ltd.**, 18 Pall Mall, London, S.W.1. Two of these which may interest automobile engineers are *Conveyor and Chain Lubrication*, and *Prevention of Screw Thread Seizure*. Each leaflet begins with general remarks on the problem, followed by recommendations for the application of the dry lubricant "dag" colloidal graphite and finally some notes are given on the different forms in which this type of material is available.

OPERATIONAL STRESSES

Measures of the Severity of Service Loading and Durability

THE following excerpts from a paper by Robert Schilling entitled "Operational Stresses in Automotive Parts" appeared in the S.A.E. Journal for May, 1951. Automotive engineers are often accused of paying little attention to stress analysis in designing parts and components, and of proceeding too much by rules of thumb and by cut and try methods in developing the strength and durability of their products. And it is quite true that the method of finding proper shapes and dimensions for highly stressed parts in automotive engineering makes more use of empirical information and less of analysis than that used in most branches of mechanical engineering. However, when we follow the steps by which we arrive at the proper design and size of our machine parts, we find that they form a sound line of reasoning.

Design practice begins with the observation of failures, either in service or in laboratory, test of complete machine parts. It leads to an empirical expression relating nominal stresses, calculated in a specific way, to the characteristics of the material.

The stresses calculated by these equations are not intended to represent true stresses, but are simply a measure of the severity of service loading and durability. Such a relation is only applicable to one type of element, manufactured in a certain manner and used in a specific type of service. In spite of these restrictions, this information permits extrapolation to applications which at first sight seem totally different. Many developments in railroad and marine transportation and designs of Ordnance vehicles have been based on passenger car practice and proved highly satisfactory.

In Table I, for example, are cited nominal stresses on the most important structural members of passenger car suspensions. This group extends over the full range of car sizes and is considered representative.

The loading conditions for which stresses are calculated do not cover all the conditions which a designer must consider. All stresses are nominal stresses as described before, without allowance for stress concentration.

Column (1) gives the stresses due to static load (five-passenger load).

Table I—Stresses in Suspension Members

		(1) Static Load Stress	(2) Lateral Load Stress	(3) Rut Stress (1)±(2)	(4) Skid Stress 2×(1)−2×(2)	(5) Overturning stress 2×(1)−2.5×(2)	(6) Bump Stress 4×(1)	(7) Brinell Hardness	(8) Yield Strength, psi
Front Wheel Spindle (Steering Knuckle)	Average	10,000	46,000	+56,000 −36,000	72,000	95,000	40,000	240–290	100,000
	High	12,000	51,000	+61,000 −41,000	82,000	107,000	48,000		
Steering Knuckle Support Upper	Average	19,000	19,500	+38,500 − 500	1,000	10,750	76,000	240–290	100,000
	High	20,500	22,000	+42,000 − 2,000	4,000	15,000	81,000		
Lower	Average	10,000	21,000	+31,000 −11,000	22,000	32,500	40,000		
	High	12,500	24,500	+37,000 −12,000	24,000	37,000	50,000		
King Pin	Average	22,000	52,000	+74,000 −30,000	60,000	86,000	88,000	Case 60 Rock.	250,000
	High	26,000	58,000	+80,000 −34,000	69,000	98,000	104,000	Core 230	95,000
Rear Axle Shaft	Average	8,000	30,000	+38,000 −22,000	44,000	59,000	32,000	330–360	145,000
	High	11,000	45,000	+52,000 −38,000	75,000	98,000	44,000		
Rear Axle Housing	Average	9,000	2,500	+11,500 − 6,500	13,000	11,750	36,000	Stamping	40,000
	High	12,500	3,500	+16,000 − 9,000	14,500	12,000	40,000		
Lower Wishbone Front Suspension	Stamping High	18,000					35,000	Stamping	40,000
	Forging High	21,000					47,000	207–255	80,000

Table II—Torsional Stress in Rear Axle Shafts Under Low Gear Torque

		Stress	Shear Yield Strength
Shear stress at smallest circular section	Average	52,000	83,000
	High	63,000	
Shear stress in spline: Torque divided by section modulus of root circle	Average	55,000	
	High	60,000	

Column (2) gives the stress due to a lateral load equal to 60 per cent of static load applied at the tire-ground contact.

Column (3) Rut stress: $=(1)\pm(2)$. This is the stress produced by bumpy or rutted roads or in fast cornering. It represents a severe loading condition which can be expected less than 100,000 times during the life of a car.

Column (4) Skid stress: $=2\times(1)-2\times(2)$. This loading consists of double static load (the whole weight of front or rear end on one side), combined with a lateral load of 60 per cent acting inwards at the tyre-ground contact. Although this loading does not represent a realistic road condition, it has been included because of its widespread use in engineering practice.

Column (5) Overturning stress: $=2\times(1)-2{\cdot}5\times(2)$. This is presented as an approximation of a more realistic overturning load. It is based on average values of tread and centre of gravity height. It also includes consideration of the fact that, due to roll, the center of gravity shifts toward the outside of the turn while the overloaded tyre under lateral load distorts and shifts its center of pressure to the inside. The lateral load is assumed partly at the tyre-ground contact and partly at the rim.

Column (6) Bump stress = stress caused by $4\times$ static load at wheel. For most parts, this stress is simply $4\times$ static load stress, but in some cases it is less because part of the overload is carried by the rubber bumper.

Column (7) The specified Brinell hardness range of the material.

Column (8) The average yield strength of the material corresponding to the lowest hardness in column (7).

The average values are not the average of all cars, but those of one car which is considered representative. The high values show the individual high for that stress; they do not all apply to the same car and hence are not calculated from the stresses shown in columns (1) and (2).

Front wheel spindle (steering knuckle). Shown are bending stresses at the fillet adjacent to the inner bearing. The rut stress runs as high as + 60 per cent, — 40 per cent of yield strength without considering stress concentration. Even with the good surface finish usual at this location, fatigue failure is hence possible. The bump stress is low, but the overturning stress reaches yield strength. Evidently this part must be designed for both static and fatigue failure.

Steering knuckle support. Bending stresses in the highest stressed sections, both above and below the kingpin boss, are given. The rut stresses are lower than for the wheel spindle, but in view of the poorer surface and less careful control of fillets, fatigue is still possible. Bump stresses and overturning stresses are safely below yield strength.

King pin. The stress is calculated for the section at the edge of the boss in which the pin is clamped; the load is assumed concentrated in the center of the bushing in which the pin rotates. The calculated stresses are hence higher than may be expected. The rut stress appears to be well below the fatigue durability of the case, and no overload stress approaches its yield point. Considering the usual case thickness of 0·040 to 0·050 in. on a diameter of one inch, the stress at the junction of case and core is about 90 per cent of the listed stress. Fatigue failure at this location is unlikely since this subsurface material has high fatigue durability, but there is only a small margin against yielding of the case under overturning or bump stress.

Rear axle shaft. Listed is the highest bending stress either at the wheel bearing or in the tapered portion. No stress is in excess of 68 per cent yield stress and, compared to the front wheel spindle, the rut stress appears low in relation to the higher yield strength of the material. However, it must be considered that the rear end may be loaded well beyond five-passenger load, and that the axle shaft, due to its rotation, is subject to more stress reversals than any part of the front suspension, so it must be designed primarily for resistance to fatigue failure.

Rear axle housing. Since the rut stresses do not reverse, the danger from fatigue is small, but bump stress approaches yield strength of material.

Lower wishbone, front suspension. High values for one design using stamped arms and one design using forged arms are shown. Fatigue loading is not calculated since it appears unimportant; the bump stress is near yield strength for the stamped part, but well below for the forged construction.

Another critical stress on axle shafts is represented by the torsion stress at the inner end, either in the splined section or in a circular section next to the splines. These stresses are listed in Table II. It can be seen that the highest nominal stress reaches 76 per cent of shear yield strength. In view of the stress concentration present, fatigue is possible even if stress reversal (in reverse gear) is infrequent.

The stress data shown in Table I and Table II check well with the service experience on these parts. That is the best reason we can have for using such calculations, although the loading conditions selected may appear arbitrary. Our aim in formulating a stress calculation must be first to predict service failure and only secondly to derive it in a rational manner. If we can do both, so much the better.

In applying this information to new designs, we must never forget its source. Although the stress equations look like generally valid relations, they fit only a narrow range of service conditions, of sizes and shapes, and of manufacturing processes. Extrapolation beyond that range is possible, but it requires care and caution.

Bright Steel Sections

AN interesting book has been published by the Exors. of James Mills, Ltd., Bredbury Steel Works, Woodley, near Stockport, dealing with their bright steel sections. It is designed particularly to illustrate those sections that are of special, probably in most cases unique, form, and the commoner outlines are not shown. The book consists almost solely of full size illustrations and the sections are very clearly visualized by being shown as white outlines on a tinted background. The dimensions are directly imposed around the sections, so that it is not becessary to refer to tabular matter. Additionally, being to scale, the outlines are in any event readily seen.

It is stated that where close dimensional accuracy is important, the sections can be drawn as close as 0.001 in. While many of the special outlines are fairly simple, others are quite complex and must constitute a technical achievement in the drawing process. Some 1400 of these special sections are shown, and the list is still being constantly augmented.

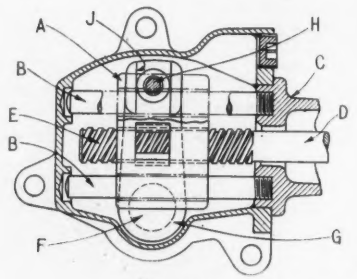
CURRENT PATENTS

A Comprehensive Review of Recent Automobile Specifications

Steering gears

IN worm and nut steering gears the nut is generally held against rotation and guided by the steering box. Consequently, the steering box must be heavily constructed and the abutting portions of the nut and box must be accurately machined.

To provide a simple steering gear permitting a relatively light and cheap steering box to be used, the nut A slides on two or more guide members B carried by the lower end of column C through which steering shaft D extends. The groove of worm E may be of constant pitch, the nut having substantial length and the central portion being removed, as shown. Alternatively, the pitch may be variable and an enshrouding sleeve type nut carrying a pin or roller engaged with the



No. 646601

groove. More than one following unit may be employed.

Rocker shaft F is rotated by pin H, secured to nut A and engaging slot J in the rocker arm G. Pin H may carry a roller or other anti-friction device. When the groove of the worm is of constant pitch, this arrangement produces an increasing value of the reduction ratio of the steering gear as the rocker arm moves away from mid-position.

Patent No. 646601. *R. Bishop and R. H. Johnston.*

Independent wheel drive

THIS invention relates specifically to cross-country vehicles having an independent drive to each independently sprung wheel. Common forms of construction have the crown wheel and the universal joint on a short transverse shaft mounted in a casing attached to the vehicle chassis, and it is usually difficult to obtain access to the crown wheel for inspection or maintenance. The invention discloses a method of construction whereby the universal joint and the crown wheel assembly can be withdrawn from the outside of the vehicle.

The main casing A is shaped to form an open ended elbow and comprises two cylindrical chambers B and C disposed normal to each other. On the exterior suitably disposed lugs are provided for attachment of the casing to the vehicle chassis D. Having a sliding fit in chamber B is a casing E with an end wall to support a bearing for the shaft F to which is splined a sleeve carrying the crown wheel G. The inner end of this shaft is supported in a bearing housed in the main casing and the outer end has an integral or attached cardan joint. A second cardan joint on the axle shaft co-operates to form a universal joint enclosed in a spherical casing H. This is supported in chamber B by two annular bearing collars J and K which are secured against rotation by dowels to casing E, which in turn is held against rotation by setscrews L. The whole assembly is axially confined by a screwed lock ring.

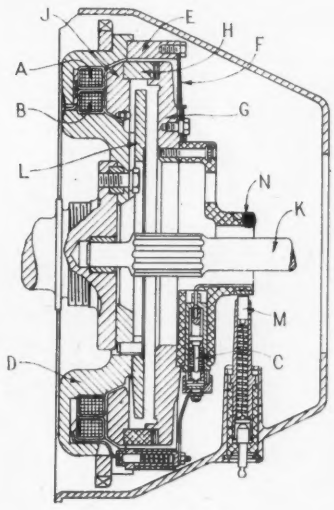
It will be seen that if the suspension arms, shown in broken outline, are detached, the wheel can be withdrawn and eventually the assembly of universal joint and crown wheel removed from the outside of the vehicle.

The bevel pinion is supported on the propeller shaft in bearings housed in chamber C. In the case of a rear wheel, as shown, the crown wheel is on the outside of the bevel pinion, but for the front wheel drive the position of the crown wheel must be reversed in order to maintain the same directional rotation. It will be appreciated, therefore, that it will be necessary to remove the pinion before the crown wheel can be withdrawn. *Patent No.* 646067. *Daimler Co. Ltd., C. M. Simpson and S. H. Shellard.*

Electro-magnetic clutch

AN electro-magnetic clutch, comprising driving and driven rotatable members and an electrical coil bringing the two members into frictional or magnetic torque-transmitting relation, does not provide sufficient slip to ensure smooth take-up of the drive at torques substantially below the maximum. In dense traffic for example, it is difficult to "inch" a motor vehicle so fitted. An improved clutch has two concentric, annular coils of which A, on being energized by generator or battery, engages the clutch to provide limited torque-transmitting capacity, and B is automatically connected in parallel with A through the medium of a centrifugal switch when the speed of the driving member exceeds a predetermined value. The switch has a spring-loaded plunger C that moves under centrifugal action to establish the connection.

In the clutch illustrated, flywheel D is attached to the engine shaft and on it is a non-magnetic flange E to which is connected, by torque-transmitting strips F of spring steel, an annular pressure plate G. The spring strips tend to disengage the clutch. Plate G is secured by an annulus

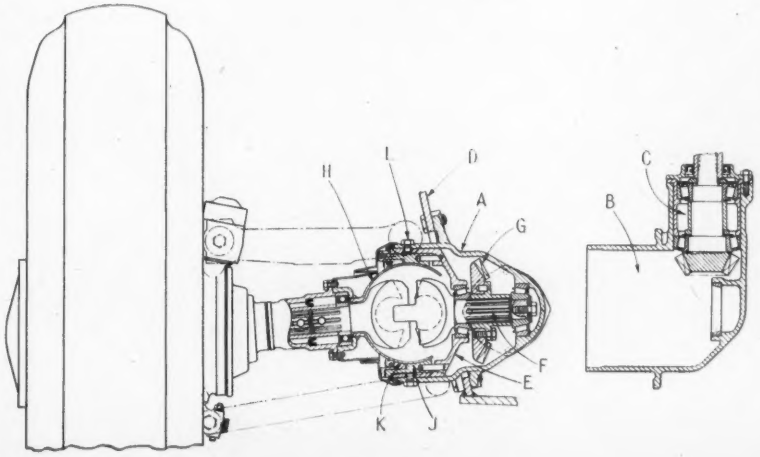


No. 645728

of non-magnetic material H to ring J constituting the armature. A conventional clutch plate (omitted in the figure) slidably splined to driven shaft K is gripped between plate G and ring L secured to the flywheel.

Electrical connection is made by a brush M, supported on the casing, through slip ring N secured to the pressure plate.

In a motor vehicle the clutch may be used in combination with an engine-driven generator applying increased potential difference to the coil A as engine speed increases from an idling speed, thereby effecting initial engagement. *Patent No.* 645728. *Humber Ltd. and A. C. Miller.*



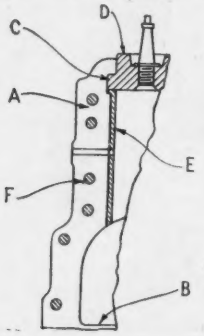
No. 646067

Engine construction

WITH the aim of securing light weight and low cost of production, it is proposed that single-cylinder engine units be constructed of half-cylinders incorporating a crankcase portion and divided on a vertical transverse plane passing through the cylinder axis. Individual units which may be either air- or water-cooled, can be bolted together to form a multi-cylinder unit.

The half sections of light-weight material are die cast to minimize the amount of machining necessary. Each section A embodies half a cylinder and half a crank chamber which is closed at the bottom by an integral sump plate B. At the head is formed an annular groove C to receive the annular flange of the cylinder head D. A steel cylinder liner E provides the working bore and also serves to secure the alignment of the die castings.

The cylinder halves are flanged and secured together by bolts F, but to achieve



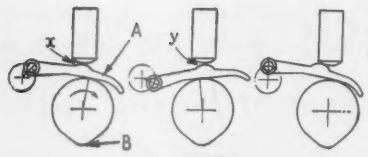
No. 646082

greater economy the halves may be united by any suitable metallic bonding process. *Patent No.* 646082. *Compact Power Products Inc.* (*U.S.A.*).

Exhaust valve gear

DELAYED closing of the exhaust valve of a two-stroke diesel engine to ensure the expulsion of the burnt charge, reduces the effective compression ratio and therefore has an adverse effect on starting from cold. This modified valve gear has interposed between cam and tappet a member that is movable laterally to advance or delay the point of closure.

The regulating member may take the form of a pivoted finger A positioned laterally and angularly by a small manually operated



No. 646404

crank. The clearance *y* in the position for starting is greater than that at *x*, in the normal running position, while the lobe B of the cam makes earlier contact with the finger. Both effects advance the closing point of the valve but, since one advances and the other retards opening, the opening point is substantially unchanged.

The third position, for use with ether, causes the valve to be open longer than normally, opening point and lift remaining substantially unchanged. If the operating lever is a bell crank, its second arm may close a butterfly valve in the air inlet pipe so as to close the valve when the mechanism is moved to the "ether" position. A strong suction is thus brought to bear upon the ether carburettor, air being induced through the pilot air inlet. A few drops of ether poured into the wick carburettor suffice to give an instant start. *Patent No.* 646404. *E. E. Pobjoy* (*D. R. Pobjoy*).

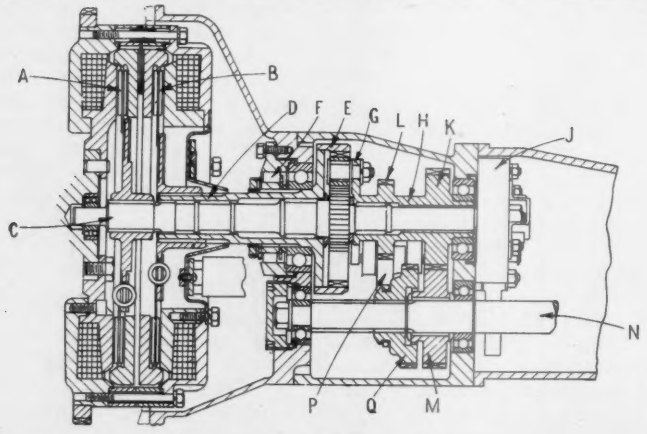
Variable-speed transmission

A mechanism comprising a pair of clutches independently operable, coupled respectively to the sun-wheel and annulus of a single epicyclic train, the planet carrier of which conveys the drive, and the sun-wheel and annulus of which are prevented from rotating in a direction opposite to that of the clutches, is the subject of this invention.

The hub of electromagnetic clutch A is splined to the sun-wheel shaft C, and the hub of clutch B is splined to sleeve D which carries, at its opposite end, annulus E. When clutch A is engaged and B free, the sun-wheel shaft is rotated and annulus E is locked by free-wheel unit F, so that planet carrier G drives the sleeve H. Conversely, when A is free and B engaged, the drive is transmitted through annulus E to planet carrier G and sleeve H, sun-wheel shaft C being locked by free-wheel unit J. Simultaneous engagement of both clutches locks the epicyclic train, and both free-wheels idle.

Sleeve H carries pinions K and L in constant mesh respectively with gear M, loosely mounted on output shaft N, and reverse idler P. A manually slidable gear Q, splined to the output shaft, engages either with M to give forward drive, or with P to give reverse. For reverse operation only clutch A is engaged.

Both free-wheels become locked if a motor vehicle equipped with this transmission stops on a rising grade. To enable the vehicle to be driven backwards without first having to be driven forwards, provision is made for the release of free-wheel J. The outer race of J is normally prevented from rotating by a roller engaging with a recess in its perpihery; manual displacement of the roller permits the whole free-wheel to rotate, and so renders its locking function inoperative. Output shaft N may be arranged for front-wheel drive, through a bevel pinion engaging a crown wheel on a tranverse shaft adjacent to the clutches. *Patent No.* 645792. *Morris Motors Ltd.*



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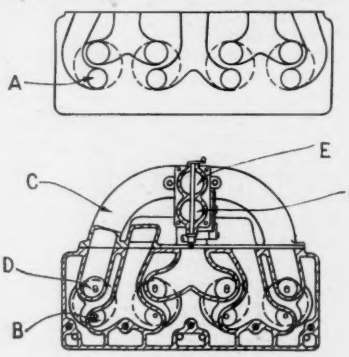
Ducting of O.H.V. engines

BY a re-arrangement of the inlet and exhaust ducts of a four-cylinder engine having overhead valves mounted in



No. 646397

planes transverse to the crankshaft the length of the cylinder head, and consequently the cylinder block, is reduced and a material saving of weight is effected. In the earlier design adjacent pairs of similar valves were siamesed and single ducts were provided for the exhaust valves A of the end cylinders. While preserving the same arrangement of valves the proposed construction is to use two siamesed ducts for the exhaust valves B.

The illustration shows a practical application. A single pipe C connects the individual ducts of the inlet valves D of the end cylinders and communicates by way of a vertical pipe (not shown) to unit E of the twin downdraught carburettor. The connection to the siamesed inlet duct for the two centre cylinders is by a horizontal pipe joining the vertical pipe, already mentioned, and the second unit F of the carburettor. Control of communication is by a flap valve which in one position connects the duct to unit E of the carburettor and in the other position to unit F. This device is interconnected with the throttle control.

Operation of the accelerator pedal through the normal range ensures that the engine is fed through the single unit E of the carburettor. Depression of the accelerator beyond this range displaces the flap valve and henceforth the feed is through both carburettor units. A spring detent device ensures that movement of the flap is sudden and positive. The driver receives a definite indication of resistance, which he only overcomes when he desires to use the maximum output of the engine. When the accelerator is released a spring returns the flap valve to the position for normal operation. *Patent No.* 646397. *E. P. Pescara* (*France*).